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A GENERAL ANALYSIS OF HIGH TEMPERATURE PIPING.

By SIDNEY AUSTIN.*

1. Introduction.

The purpose of this paper is to present a general set of equations and to intregate them with previously published work to form a method of analysis of high temperature piping for expansion stresses, moments, forces and movements as used at the New York Navy Yard. This method has been in use for some time and has given satisfaction in the solution of all problems that have come up. The general equations to be derived have proved to be satisfactory in the solution of the multi-anchor problem and in the solution of problems containing all the types of restraints that may be imposed upon a system of piping.

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The general equations have been developed to make full use of the "Meyer-Hovgaard" method of analysis. It may be looked upon as an extension of the method previously printed[1]. For speed in actual operation, in lieu of the section forms as given in the above paper, it has been found more convenient and satisfactory to use the section forms that were published in a subsequent paper[2].

Numerous articles on this subject have appeared in the past ten or fifteen years. The "Design of Piping Systems"[3] includes as complete a bibliography on the subject as is known to exist. This reference[3] develops a complete and exact solution of the problem. It is to be pointed out, however, that the general equations to be given in this paper were developed purely from the point of view of extending the "Meyer-Hovgaard" method to the solution of the more complex problems of multi-branch and vari-anchored piping systems. This became necessary when modern piping lay-outs being developed became incapable of direct analysis without approximating the runs of piping into two-anchor problems in order to be able to handle the analysis by the simpler "Meyer-Hovgaard" method then in use[1].

2. General Description.

The problem of expansion forces and moments in piping systems can be considered as a variation of the case of a complex indeterminate beam. In the case of a beam one is usually given the loading, and analysis is required to determine the moments (and consequently the stresses) and the deflection of points on the beam. In contradistinction, for the case of the piping problem one is given the deflections (the thermal expansions that take place and deflect the piping) and analysis is required to give the forces and moments that arise due to the deflections.

The statement of the simple problem may best be demonstrated by referring to Figure I. Given a single-plane piping arrangement fixed at each end, it may be orientated in an orthogonal coordinate system as shown, so that one end is placed at the origin, 0. The far end of the piping, A, is assumed permanently fixed, and if the end at the origin is freed it will undergo movements contrary to the positive direction of the X- and Y-axes due to the thermal expansion and reach the new position 0′, as indicated by the dotted line. To return the end of the pipe at O′ back to the origin 0, the external forces P and Q acting in

the X- and Y-directions, respectively, must be exerted, and to maintain end conditions of fixity, the moment M_0 in the X-Y plane must be applied to return the slope of the piping at this end to its original value. The piping then assumes a configuration indicated by the dot-dash line. Once the moment and forces, the so-called end-reactions, at the origin are known, the moment at any point in the pipe may be found, and, consequently, the stresses may be calculated.

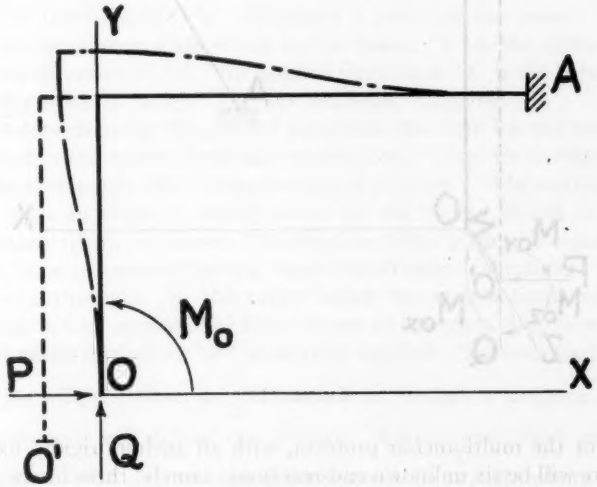


FIGURE I.

The solution of the problem thus requires finding three unknowns, P, Q and M_0. Generally speaking this requires three independent equations. For this case these equations would arise: one from the deflection in the X-direction, the second from the deflection in the Y-direction, and the third from slope considerations.

The more general problem in three-dimensional space can also be demonstrated by referring to a diagram, Figure II. To return the end of the piping at 0′ to the origin 0, three forces, P, Q and N, must be exerted, and to maintain conditions of fixity, three moments, M_{0x}, M_{0y} and M_{0z}, must be applied. There are thus six unknowns and the solution will require six

independent equations. Three of these are obtained from the consideration of the deflections along each of the three axes, X, Y and Z, and the remaining three from slope considerations in each of the three planes, X-Y, Y-Z and Z-X.

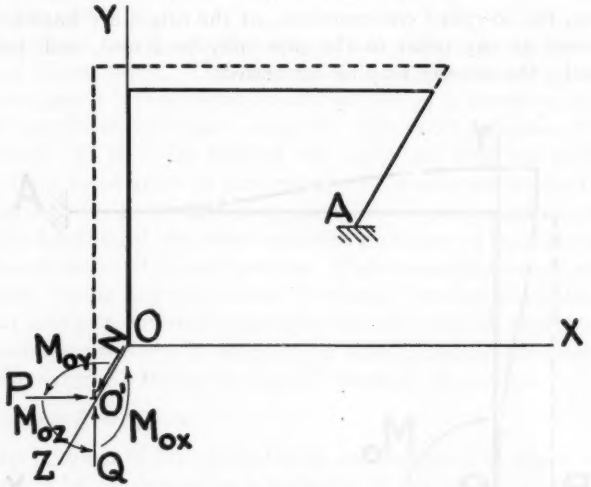


FIGURE II.

For the multi-anchor problem, with all anchors rigidly fixed, there will be six unknown end-reactions, namely, three forces and three moments, at each of the anchors. However, if the values for the end-reactions at all the anchors but one are known, then the values for the last anchor can be found from the conditions of equilibrium, namely, that the summation of forces and moments at any one point are equal to zero. Thus a system with "n" complete anchors will require a solution at (n-1) anchors, or 6 (n-1) equations will be necessary for simultaneous solution, and the last anchor is then easily determined from conditions of equilibrium.

One of the more powerful methods of solution for indeterminate beams is to be found in Castigliano's Theorem. This theorem states that the rate of change of the total energy which is stored up in a system with respect to a force applied at a point is equal to the deflection of this point in the direction of the applied force.

Similarly, the rate of change of energy with respect to a moment applied at this point is equal to the rotation at this point in the plane in which the moment acts. Mathematically, this may be stated as:

$$S_a = \frac{\partial W}{\partial F_a} \tag{1.}$$

$$\phi_a = \frac{\partial W}{\partial M_a} \tag{2.}$$

where the subscript "a" represents a point on the beam; "W" is the total energy stored up in the beam; "S" is the deflection in the direction of "F," the applied force; and "ϕ" is the rotation in the plane in which "M," the moment, is applied.

Before deriving the general equations the term for the energy stored in the system deserves examination. Consider an element of pipe of length "dL" along the axis of the pipe. The work done on such an element, which would be the energy stored in the element (if no permanent deformation takes place), is equal to the force or moment acting times the distance or rotation that it acts through. In the range below the elastic limit, where Hooke's Law applies, the force is zero at the start of application and varies linearly to "F" when fully applied. The average force acting will, therefore, be $\frac{1}{2}$F, which acts through a distance that the force F will deflect the element. For a direct force acting axially on the element, the deflection will be $\frac{F}{EA}$ dL, where "E" is Young's modulus, and "A" is the cross-sectional area. The energy in such a case will then be

$$dW = \frac{F^2}{2EA} dL \tag{3.}$$

For the case of a shear force, the deflection would be $f\frac{F_s}{GA}$ dL, where "Fs" is the shear force, "f" is a form factor (= 2 for a thin cylinder), and "G" is the shearing modulus (= E/2.6 for steel with Poisson's ratio = 0.3). The energy for a shear force on the element then becomes,

$$dW = f\frac{F_s^2}{2GA} dL = 5.2\frac{F_s^2}{2EA} dL \text{ (for steel pipe)} \tag{3a.}$$

For a bending moment acting on the element, the rotation will be M dL/EI, where "M" is the bending moment, and "I" is the moment of inertia of the pipe section about a diameter. The energy is

$$dW = \frac{M^2}{2EI}\,dL \tag{4.}$$

And for a torsional moment this becomes

$$dW = \frac{M_t^{\,2}}{2GJ}\,dL = 1.3\,\frac{M_t^{\,2}}{2EI}\,dL \text{ (for steel pipe)} \tag{4a.}$$

where "M_t" is the torsional moment, and "J" is the polar moment of inertia and equals 2I for a cylinder.

If factors "k" are now introduced to take care of factors of proportionality for G, J, and f above, and to take into account increased flexibility for bending in the plane of a bend and effects due to corrugated or creased piping, the energy equation may be written as:

$$W = \sum \int \frac{k\,M^2}{2\,EI}\,dL + \sum \int \frac{k\,F^2}{2\,EA}\,dL \tag{5.}$$

where "W" is now the total energy stored in the piping system between the limits of integration.

Examining the mechanics by which deflection takes place in a piping system, part of this will be caused by flexure due to the moments, and part by the deflections caused by the direct forces. In the case of a beam it is known that by far the largest part of the deflection is due to flexure, and the effects of the direct forces are normally disregarded as of a secondary nature. Following through with this reasoning, terms due to equations (3) and (3a) will be small compared with those due to (4) and (4a). The "Meyer-Hovgaard" method neglects these effects. Since this paper is actually an extension of that method, and since the effects of the direct forces are of little consequence in most engineering solutions, they shall also be neglected in the derivation of the general equations. The assumption is made, therefore, that to a sufficient degree of accuracy the energy in a piping system can be written,

$$W = \sum \int \frac{kM^2}{2\,EI}\,dL. \tag{6.}$$

(Occasionally problems arise where the effects of the direct forces are not negligible. Such would be the case where the piping is rather inflexible and, therefore, the forces imposed on the piping are high. However, such piping is rare in shipboard use.)

3. The General Equations.

Nomenclature:

P, Q, N = Forces, positive in the x, y, and z plus directions, respectively.

M_x, M_y, M_z = Moments in the planes perpendicular to x, y, and z, respectively, signs governed by vectorial presentation.

ϕ_x, ϕ_y, ϕ_z = Rotations in these planes, signs similarly fixed.

S_x, S_y, S_z = Deflections along the coordinate axes.

W = Energy stored in the system, or work done on the system.

M' = Internal moment.

0 = Subscript for origin of coordinate system.

A, B, etc. = Subscripts for points on the pipe.

k = Flexibility factor for bends or creased piping.

= Proportionality factor for torsion, or combined torsion and bending.

E = Young's modulus.

I = Moment of inertia of pipe section about a diameter.

dL = Element of length along axis of pipe.

Δ = Finite difference.

δ = Finite difference.

c = Subscript for considered point on pipe.

i = Subscript for initial point on pipe.

Δx = Thermal expansion between initial and considered points, in the x-direction.

$= \tau (\bar{x}_c - \bar{x}_i)$, where "$\tau$" is the thermal expansion per unit length over the temperature range for the particular case.

δx = Difference in actual movement of considered point above that of initial point, in the x-direction.

$= m_{xc} - m_{xi}$, where "m" is the actual externally applied movement.

$\bar{x}, \bar{y}, \bar{z}$ = Cartesian coördinates of considered and initial points.

$\Delta\phi = \phi_c - \phi_i$ = Difference in angle of rotation between considered point and initial point, respectively.

$\Delta(\bar{x}\phi_z) = \bar{x}_c\phi_{zc} - \bar{x}_i\phi_{zi}$.

= Difference in rotation effects at considered point and initial point, respectively, in the y-direction, due to rotation in the X-Y plane.

$\Delta(\bar{y}\phi_z)$ = Same in x-direction due to rotation in X-Y plane.

$\Delta(\bar{x}\phi_y)$ = Same in z-direction due to rotation in Z-X plane.

$\Delta(\bar{z}\phi_y)$ = Same in x-direction due to rotation in Z-X plane.

$\Delta(\bar{y}\phi_x)$ = Same in z-direction due to rotation in Y-Z plane.

$\Delta(\bar{z}\phi_x)$ = Same in y-direction due to rotation in Y-Z plane.

$A_s, A_x, A_y,$
A_{x^2}, A_{y^2}, A_{xy} = Meyer-Hovgaard coefficients for X-Y plane, see text.

$B_s, B_y, B_z,$
B_{y^2}, B_{z^2}, B_{yz} = Meyer-Hovgaard coefficients for Y-Z plane.

$C_s, C_z, C_x,$
C_{z^2}, C_{x^2}, C_{zx} = Meyer-Hovgaard coefficients for Z-X plane.

Any force and moment, or system of forces and moments, at a point in space may be resolved into a system of three forces and three moments along any set of orthogonal coordinate axes. Such a system is shown in Figure III, with all forces and moments presented in their positive sense.

Having chosen a run of piping, let it be orientated in an orthogonal right-hand coordinate system so that an anchor,

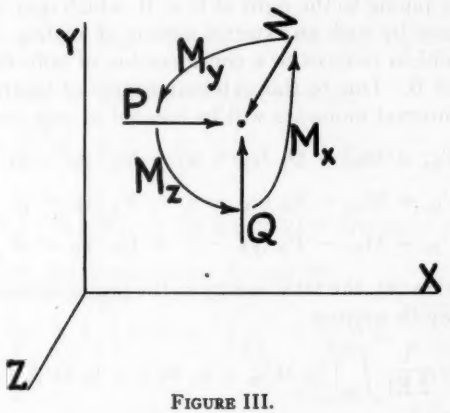


FIGURE III.

assumed completely rigid, is placed at the origin, Figure IV. Let P be any point on the pipe, with coordinates x, y and z, that lies between O and B. Let B be a chosen point on the pipe, determined only so that it lies on a run of pipe that has no restraints or junction points between it and the origin. Let a system of external forces and moments, such as shown in Figure III, be placed at the point B in Figure IV. This system of forces and moments may be externally applied at B if no restraints are assumed beyond B, or may be considered as the accumulated

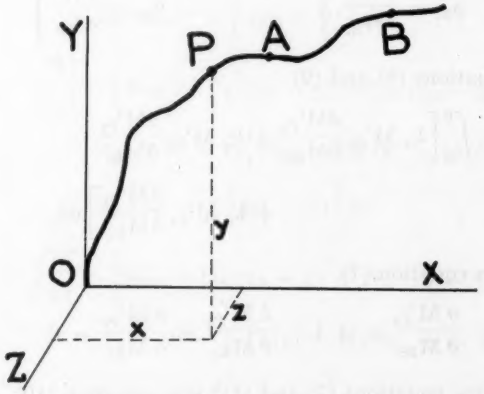


FIGURE IV.

effect of any piping to the right of B at B, which may be equivalently replaced by such an external system of loading. In either case the problem reduces to a consideration of only the run between O and B. Due to this external system of loading at B, a system of internal moments will be created at any point P.

$$\left.\begin{aligned} M'_{Px} &= M_{Bx} - Q_B (z_B - z) + N_B (y_B - y) \\ M'_{Py} &= M_{By} - N_B (x_B - x) + P_B (z_B - z) \\ M'_{Pz} &= M_{Bz} - P_B (y_B - y) + Q_B (x_B - x) \end{aligned}\right\} \qquad (7.)$$

Using equation (6), the total energy in the piping system between O and B may be written

$$W = \frac{1}{2\,EI} \int_0^B \left[k_x M'^2_{Px} + k_y M'^2_{Py} + k_z M'^2_{Pz} \right] dL \qquad (8.)$$

where E and I are assumed constant, i. e., the piping is of uniform material and dimension.

Making use of Castigliano's Theorem, equations (1) and (2), the following six equations may be written:

$$\left.\begin{aligned} \phi_{Bx} &= \frac{\partial W}{\partial M_{Bx}} \\ \phi_{By} &= \frac{\partial W}{\partial M_{By}} \\ \phi_{Bz} &= \frac{\partial W}{\partial M_{Bz}} \end{aligned}\right\} \qquad \left.\begin{aligned} S_{Bx} &= \frac{\partial W}{\partial P_B} \\ S_{By} &= \frac{\partial W}{\partial Q_B} \\ S_{Bz} &= \frac{\partial W}{\partial N_B} \end{aligned}\right\} \qquad (9.)$$

Using equations (8) and (9)

$$EI\,\phi_{Bx} = \int_0^B \left[k_x M'_{Px} \frac{\partial M'_{Px}}{\partial M_{Bx}} + k_y M'_{Py} \frac{\partial M'_{Py}}{\partial M_{Bx}} + k_z M'_{Pz} \frac{\partial M'_{Pz}}{\partial M_{Bx}} \right] dL \qquad (10.)$$

Now from equation (7)

$$\frac{\partial M'_{Px}}{\partial M_{Bx}} = +1 \; ; \quad \frac{\partial M'_{Py}}{\partial M_{Bx}} = \frac{\partial M'_{Pz}}{\partial M_{Bx}} = 0. \qquad (11.)$$

Substituting equations (7) and (11) into equation (10),

$$\left.\begin{aligned} EI\,\phi_{Bx} &= \int_0^B k_x\Big[M_{Bx} - Q_B\,(z_B - z) + N_B\,(y_B - y)\Big]dL \\ \text{and similarly for the other rotation terms,} & \\ EI\,\phi_{By} &= \int_0^B k_y\Big[M_{By} - N_B\,(x_B - x) + P_B\,(z_B - z)\Big]dL \\ EI\,\phi_{Bz} &= \int_0^B k_z\Big[M_{Bz} - P_B\,(y_B - y) + Q_B\,(x_B - x)\Big]dL \end{aligned}\right\} \quad (12.)$$

Similarly, using equations (8) and (9),

$$EI\,S_{Bx} = \int_0^B \Big[k_x\,M'_{Px}\frac{\partial\,M'_{Px}}{\partial\,P_B} + k_y\,M'_{Py}\frac{\partial\,M'_{Py}}{\partial\,P_B} + k_z\,M'_{Pz}\frac{\partial\,M'_{Pz}}{\partial\,P_B}\Big]dL. \quad (13.)$$

And again from equation (7)

$$\frac{\partial\,M'_{Px}}{\partial\,P_B} = 0;\ \frac{\partial\,M'_{Py}}{\partial\,P_B} = +\,(Z_B - Z);\ \frac{\partial\,M'_{Pz}}{\partial\,P_B} = -\,(y_B - y). \quad (14.)$$

Substituting in equation (13), and proceeding in the same way for the remaining terms

$$\left.\begin{aligned} EI\,S_{Bx} &= \int_0^B \Big[k_y\Big\{M_{By} - N_B\,(x_B - x) + P_B\,(z_B - z)\Big\}(z_B - z) - k_z\Big\{M_{Bz} - P_B\,(y_B - y) + Q_B\,(x_B - x)\Big\}(y_B - y)\Big]dL \\ EI\,S_{By} &= \int_0^B \Big[k_z\Big\{M_{Bz} - P_B\,(y_B - y) + Q_B\,(x_B - x)\Big\}(x_B - x) - k_x\Big\{M_{Bx} - Q_B\,(z - z) + N_B\,(y_B - y)\Big\}(z_B - z)\Big]dL \\ EI\,S_{Bz} &= \int_0^B \Big[k_x\Big\{M_{Bx} - Q_B\,(z_B - z) + N_B\,(y_B - y)\Big\}(y_B - y) - k_y\Big\{M_{By} - N_B\,(x_B - x) + P_B\,(z_B - z)\Big\}(x_B - x)\Big]dL \end{aligned}\right\} \quad (15.)$$

Equations (12) and (15) now express the rotations and deflections at B in terms of the system of external forces and moments at B. It is desirous to express these equations in terms of the external forces and moments at the origin. To accomplish this, the following transformations must be made. From the first of equations (7),

$$M'_{0x} = M_{Bx} - Q_B z_B + N_B y_B \tag{16.}$$

The relations between the internal forces and moments and the external ones at O are:

$$\left.\begin{aligned} M_{0x} &= - M'_{0x} \\ Q_0 &= - Q_B \\ N_0 &= - N_B \end{aligned}\right\} \tag{17.}$$

Therefore, equation (16) may be rewritten as

$$\left.\begin{aligned} M_{Bx} &= - M_{0x} - Q_0 z_B + N_0 y_B \\ \text{And similarly,} \\ M_{By} &= - M_{0y} - N_0 x_B + P_0 z_B \\ M_{Bz} &= - M_{0z} - P_0 y_B + Q_0 x_B \end{aligned}\right\} \tag{18.}$$

Substituting (18) back into (12), the latter become

$$\left.\begin{aligned} - EI\, \phi_{Bx} &= \int_0^B k_x \Big[M_{0x} + Q_0 z - N_0 y \Big] dL \\ - EI\, \phi_{By} &= \int_0^B k_y \Big[M_{0y} + N_0 x - P_0 z \Big] dL \\ - EI\, \phi_{Bz} &= \int_0^B k_z \Big[M_{0z} + P_0 y - Q_0 x \Big] dL \end{aligned}\right\} \tag{19.}$$

Similarly, substituting back into equations (15), equations (15) become

$$
\left.\begin{aligned}
-EI\,S_{Bx} &= \int_0^B \Big[k_y \Big\{ M_{0y} + N_0 x - P_0 z \Big\}(z_B - z) - k_z \\
&\qquad \Big\{ M_{0z} + P_0 y - Q_0 x \Big\}(y_B - y) \Big] dL \\
-EI\,S_{By} &= \int_0^B \Big[k_z \Big\{ M_{0z} + P_0 y - Q_0 x \Big\}(x_B - x) - k_x \\
&\qquad \Big\{ M_{0x} + Q_0 z - N_0 y \Big\}(z_B - z) \Big] dL \\
-EI\,S_{Bz} &= \int_0^B \Big[k_x \Big\{ M_{0x} + Q_0 z - N_0 y \Big\}(y_B - y) - k_y \\
&\qquad \Big\{ M_{0y} + N_0 x - P_0 z \Big\}(x_B - x) \Big] dL
\end{aligned}\right\} \quad (20.)
$$

Now to the first of equations (20) add z_B (EI ϕ_{By}) and subtract y_B (EI ϕ_{Bz}). To the second equation add x_B (EI ϕ_{Bz}) and subtract z_B (EI ϕ_{Bx}). To the last equation add y_B (EI ϕ_{Bx}) and subtract x_B(EIϕ_{By}). There now results

$$
\left.\begin{aligned}
EI\Big[-S_{Bx} - y_B \phi_{Bz} + z_B \phi_{By} \Big] &= \int_0^B \Big[k_z \Big\{ M_{0z} + P_0 y \\
&\qquad - Q_0 x \Big\} y - k_y \Big\{ M_{0y} + N_0 x - P_0 z \Big\} z \Big] dL \\
EI\Big[-S_{By} - z_B \phi_{Bx} + x_B \phi_{Bz} \Big] &= \int_0^B \Big[k_x \Big\{ M_{0x} + Q_0 z \\
&\qquad - N_0 y \Big\} z - k_z \Big\{ M_{0z} + P_0 y - Q_0 x \Big\} x \Big] dL \\
EI\Big[-S_{Bz} - x_B \phi_{By} + y_B \phi_{Bx} \Big] &= \int_0^B \Big[k_y \Big\{ M_{0y} + N_0 x \\
&\qquad - P_0 z \Big\} x - k_x \Big\{ M_{0x} + Q_0 z - N_0 y \Big\} y \Big] dL
\end{aligned}\right\} \quad (21.)
$$

Now for the point A in Figure IV, the following equations can be written by following a derivation similar to the preceding one. Since the pipe is in equilibrium, the system of external loading to be applied at A is one that is in equilibrium with the previous one, or determined so that the external loading at O is identical with that of the previous case.

The rotation equations similar to equations (19) then become,

$$\left.\begin{aligned} -EI\,\phi_{Ax} &= \int_0^A k_x\Big[M_{0x} + Q_0 z - N_0 y\Big]dL \\ -EI\,\phi_{Ay} &= \int_0^A k_y\Big[M_{0y} + N_0 x - P_0 z\Big]dL \\ -EI\,\phi_{Az} &= \int_0^A k_z\Big[M_{0z} + P_0 y - Q_0 x\Big]dL \end{aligned}\right\} \quad (22.)$$

And the deflection equations similar to equations (21) are,

$$\left.\begin{aligned} EI\Big[-S_{Ax} - y_A\,\phi_{Az} + z_A\,\phi_{Ay}\Big] &= \int_0^A\Big[k_z\Big\{M_{0z} + P_0 y - Q_0 x\Big\}y - k_y\Big\{M_{0y} + N_0 x - P_0 z\Big\}z\Big]dL \\ EI\Big[-S_{Ay} - z_A\,\phi_{Ax} + x_A\,\phi_{Az}\Big] &= \int_0^A\Big[k_x\Big\{M_{0x} + Q_0 z - N_0 y\Big\}z - k_z\Big\{M_{0z} + P_0 y - Q_0 x\Big\}x\Big]dL \\ EI\Big[-S_{Az} - x_A\,\phi_{Ay} + y_A\,\phi_{Ax}\Big] &= \int_0^A\Big[k_y\Big\{M_{0y} + N_0 x - P_0 z\Big\}x - k_x\Big\{M_{0x} + Q_0 z - N_0 y\Big\}y\Big]dL \end{aligned}\right\} \quad (23.)$$

If equations (22) are subtracted from (19), and equations (23) from (21), there now results the general equations between A and B, any two points in a piping system, without any type of external restraint between them, being acted upon by an external system of forces and moments that has been transferred to the origin.

Before doing this, consider some of the terms on the left side of the equations with the point of view of changing the symbolism slightly. Now ϕ_{Bx} is the absolute value of the rotation at B, and similarly ϕ_{Ax} at A. Now if B be called the considered point, and A the initial point, in terms of the nomenclature set up, it may be said that $\phi_{Bx} - \phi_{Ax} = \Delta\phi_x$. Also $\Delta(\bar{y}\,\phi_z)$ may be used for $y_B\,\phi_{Bz} - y_A\,\phi_{Az}$, etc. The term S_{Bx} is the deflection along the x-axis at B due to the loading applied there, and is positive in the plus x-direction. The thermal expansion in the x-direction

from the anchor at 0 will be positive in the same sense. The deflection in the piping due to this expansion—thermal deflection—is always opposite in sign to the thermal expansion. There may also be external movements at the anchors, and these will have a positive direction in the same sense as S_x. If Δx is defined as the thermal expansion, so that the thermal deflection is $-\Delta x$, between points A and B, and δx is the difference in external movements between B and A, then it may be said that $S_{Bx} - S_{Ax} = \Delta S_x = -\Delta x + \delta x$. Making one further change in notation as far as forces are concerned, since the forces are always to be considered as external forces at the origin, we can drop the subscripts.

The general equations thus become:

$$\left.\begin{aligned} -EI\,\Delta\phi_x &= \int_A^B k_x\Big[M_{0x} + Q_z - N_y\Big]dL \\ -EI\,\Delta\phi_y &= \int_A^B k_y\Big[M_{0y} + N_x - P_z\Big]dL \\ -EI\,\Delta\phi_z &= \int_A^B k_z\Big[M_{0z} + P_y - Q_x\Big]dL \end{aligned}\right\} \quad (24.)$$

$$\left.\begin{aligned} EI\Big[\Delta x - \delta x - \Delta(\bar{y}\phi_z) + \Delta(\bar{z}\phi_y)\Big] &= \int_A^B\Big[k_z\Big\{M_{0z} + Py - Qx\Big\}y - k_y\Big\{M_{0y} + Nx - Pz\Big\}z\Big]dL \\ EI\Big[\Delta y - \delta y - \Delta(\bar{z}\phi_x) + \Delta(\bar{x}\phi_z)\Big] &= \int_A^B\Big[k_x\Big\{M_{0x} + Qz - Ny\Big\}z - k_z\Big\{M_{0z} + Py - Qx\Big\}x\Big]dL \\ EI\Big[\Delta z - \delta z - \Delta(\bar{x}\phi_y) + \Delta(\bar{y}\phi_x)\Big] &= \int_A^B\Big[k_y\Big\{M_{0y} + Nx - Pz\Big\}x - k_x\Big\{M_{0x} + Qz - Ny\Big\}y\Big]dL \end{aligned}\right\} \quad (25.)$$

4. Introduction of Section Forms.

The coefficients of the unknowns in the general equations are of the form

$$\text{Coefficient} = \int_A^B k\, f(x, y, z)\, dL \quad (26.)$$

Suppose now that the pipe is broken up into "n" convenient sections, each section being taken as a straight length of pipe, a bend, or a portion of a bend, etc. The above integral can then be broken up into

$$C = \int_{A}^{P_1} k_1 f_1 \, dL + \int_{P_1}^{P_2} k_2 f_2 \, dL + \text{------} + \int_{P_{n-2}}^{P_{n-1}} k_{n-1} f_{n-1} \, dL + \int_{P_{n-1}}^{B} k_n f_n \, dL \qquad (27.)$$

The individual integrals then depend only on the geometry of the section of pipe being integrated over—since the integral is a function only of the length and the coordinates of the individual section, and the factor "k" which is a function only of the angular position in space of the section. Equation (27) can then be rewritten as

$$C = C_1 + C_2 + C_3 + \text{----------} + C_{n-1} + C_n = \sum_{n-1}^{n} C_1. \qquad (28.)$$

The individual coefficients can be tabulated on forms as given in other papers[1, 2], and then summed up for the entire pipe. Each section of pipe gives rise to six coefficients in each of the three planes. The summation of these terms are the coefficients to be used in the equations.

In terms of the symbols used in the Meyer-Hovgaard method[1], the following are equivalents:

$$\left.\begin{array}{lll} \int k_z \, dL = A_s & \int k_z \, x \, dL = A_x & \int k_z \, y \, dL = A_y \\ \int k_x \, dL = B_s & \int k_x \, y \, dL = B_y & \int k_x \, z \, dL = B_z \\ \int k_y \, dL = C_s & \int k_y \, z \, dL = C_z & \int k_y \, x \, dL = C_x \\ \int k_z \, x^2 \, dL = A_{x^2} & \int k_z \, y^2 \, dL = A_{y^2} & \int k_z \, xy \, dL = A_{xy} \\ \int k_x \, y^2 \, dL = B_{y^2} & \int k_x \, z^2 \, dL = B_{z^2} & \int k_x \, yz \, dL = B_{yz} \\ \int k_y \, z^2 \, dL = C_{z^2} & \int k_y \, x^2 \, dL = C_{x^2} & \int k_y \, zx \, dL = C_{zx} \end{array}\right\} \qquad (29.)$$

Using these symbols, the general equations (24) and (25) become,

$$
\left.
\begin{aligned}
-EI\Delta\phi_x = {} & B_s M_{0x} \\
& +B_z Q \qquad -B_y N \\
-EI\Delta\phi_y = {} & C_s M_{0y} \\
& -C_z P \qquad +C_x N \\
-EI\Delta\phi_z = {} & \\
& A_s M_{0z} + A_y P \qquad -A_x Q \\
EI\,[\Delta x - \delta x - \Delta(\bar{y}\phi_z) + \Delta(\bar{z}\phi_y)] = {} & -C_z M_{0y} \\
& +A_y M_{0z} + (A_{y^2} + C_{z^2}) P - A_{xy} Q \qquad -C_{zx} N \\
EI\,[\Delta y - \delta y - \Delta(\bar{z}\phi_x) + \Delta(\bar{x}\phi_z)] = {} & B_z M_{0x} \\
& -A_x M_{0z} - A_{xy} P \qquad +(A_{x^2} + B_{z^2}) Q - B_{yz} N \\
EI\,[\Delta z - \delta z - \Delta(\bar{x}\phi_y) + \Delta(\bar{y}\phi_x)] = {} & -B_y M_{0x} + C_x M_{0y} \\
& -C_{zx} P \qquad -B_{yz} Q \qquad +(B_{y^2} + C_{x^2}) N
\end{aligned}
\right\} \quad (30.)
$$

For convenience, at this point, the definition of pertinent terms on the left of these equations are repeated.

$\Delta\phi$ = Difference in angle of rotation between considered point and initial point, respectively $= \phi_c - \phi_i$.

ϕ_z, etc. = Rotation in X-Y plane, etc.

Δx, etc. = Thermal expansion between initial point and considered point, in the x-direction, etc. $= \tau(\bar{x}_c - \bar{x}_i)$.

δx, etc. = Difference in actual movement of considered point above that of initial point, in the x-direction, etc. $= m_{xc} - m_{xi}$.

$\bar{x}$, $\bar{y}$, $\bar{z}$ = Cartesian coordinates of considered point and initial point.

$\Delta(\bar{x}\phi_z)$, etc. = Difference in rotation effects at considered point and initial point, respectively, in the y-direction due to the X-Y plane, etc. $= \bar{x}_c\phi_{zc} - \bar{x}_i\phi_{zi}$.

5. The Equations for a Single Plane Pipe.

The general equations for a single plane pipe can easily be found by reduction from equations (30). For a pipe to be considered as a single plane pipe, in addition to lying in a single plane, all disarrangements (Δz, δz, $\Delta\phi_y$, $\Delta\phi_x$) perpendicular to the plane of the piping must be zero. If the pipe be placed in an X-Y plane at $z = 0$, the shape coefficients B_z, B_{z^2}, B_{yz}, C_z, C_{z^2}, and C_{zx} will become zero. The equations (30) then reduce to two independent sets of three equations each.
One set is,

$$\left.\begin{aligned} A_s M_{0z} + A_y P - A_x Q &= -EI\,\Delta\phi_z \\ A_y M_{0z} + A_{y^2} P - A_{xy} Q &= EI\,[\Delta x - \delta x - \Delta(\bar{y}\phi_z)] \\ -A_x M_{0z} - A_{xy} P + A_{x^2} Q &= EI\,[\Delta y - \delta y + \Delta(\bar{x}\phi_z)] \end{aligned}\right\} \quad (31.)$$

These may be solved for M_{0z}, P and Q. The second set is,

$$\left.\begin{aligned} B_s M_{0x} - B_y N &= 0 \\ C_s M_{0y} + C_x N &= 0 \\ -B_y M_{0z} + C_x M_{0y} + (B_{y^2} + C_{x^2}) N &= 0 \end{aligned}\right\} \quad (32.)$$

This set gives rise to the solution $M_{0x} = M_{0y} = N = 0$.

For the conventional single plane pipe with two fixed anchors, equations (31) reduce to,

$$\left.\begin{aligned} A_s M_{0z} + A_y P - A_x Q &= 0 \\ A_y M_{0z} + A_{y^2} P - A_{xy} Q &= EI\,\Delta x \\ -A_x M_{0z} - A_{xy} P + A_{x^2} Q &= EI\,\Delta y \end{aligned}\right\} \quad (33.)$$

This is the form that the equations take for a simple single plane pipe in the Meyer-Hovgaard method[1].

6. The Three-Plane Two-Anchor Problem.

In this case the anchors are again assumed fixed so that the expressions for rotation are equal to zero. This gives rise to the three rotation equations

$$\left.\begin{aligned} A_s M_{0z} + A_y P - A_x Q &= 0 \\ B_s M_{0x} + B_z Q - B_y N &= 0 \\ C_s M_{0y} + C_x N - C_z P &= 0 \end{aligned}\right\} \quad (34.)$$

These equations can be solved for the moments in terms of two forces, since each equation contains only one moment. If these moments are then substituted for in the deflection equations, the deflection equations become,

$$\left.\begin{aligned} &[A_{y^2} + C_{z^2} - (A_y)^2/A_s - (C_z)^2/C_s]\,P - [A_{xy} - A_xA_y/A_s]\,Q - [C_{zx} - C_zC_x/C_s]\,N = EI\,\Delta x \\ &-[A_{xy} - A_xA_y/A_s]\,P + [A_{x^2} + B_{z^2} - (A_x)^2/A_s - (B_z)^2/B_s]\,Q - [B_{yz} - B_yB_z/B_s]\,N = EI\,\Delta y \\ &-[C_{zx} - C_zC_x/C_s]\,P - [B_{yz} - B_yB_z/B_s]\,Q + [C_{x^2} + B_{y^2} - (C_x)^2/C_s - (B_y)^2/B_s]N = EI\,\Delta z \end{aligned}\right\} \quad (35.)$$

Now using the α, β and λ terms introduced in the original Meyer-Hovgaard presentation[1], these equations become,

$$\left.\begin{aligned} \beta_3\,P - \beta_2\,Q - \beta_1\,N &= EI\,\Delta x \\ -\beta_2\,P + \lambda_2\,Q - \alpha_2\,N &= EI\,\Delta y \\ -\beta_1\,P - \alpha_2\,Q + \alpha_1\,N &= EI\,\Delta z \end{aligned}\right\} \quad (36.)$$

This reduces the six equations to the solution of only three simultaneous equations for the forces. The moments are then solved for independently by using the values for the forces found from equations (36) in equations (34).

7. The Factor "k" for Combined Bending and Torsion.

For purposes of checking the section forms[1, 2] by means of equations (29), additional comment is necessary as to what to use for the k-factors. This is particularly so where the section is so placed as to be under the combined action of bending and torsion by certain of the moments.

In developing the general equations, the energy in the system has been defined by equation (8). For plain piping, i. e. not corrugated or otherwise treated to increase its flexibility, k_b is unity for bending, or equal The Von Karman flexibility factor for bending in the plane of curvature of a curved section, and k_t is 1.3 for torsion. It now remains to examine the form of the equivalent k for the case of combined bending and torsion.

Suppose an element of plain pipe be located in a plane parallel to the X-Y plane. The internal moment M'_z then acts only in

pure bending, but the internal moments M'_x and M'_y may act in both bending and torsion. Let such an element be represented in the accompanying diagram, Figure V. Then M'_x may be resolved into two components parallel and perpendicular to the element, M'_{xt} in torsion and M'_{xb} in bending, respectively. The same may be done for M'_y. Then the following relations hold:

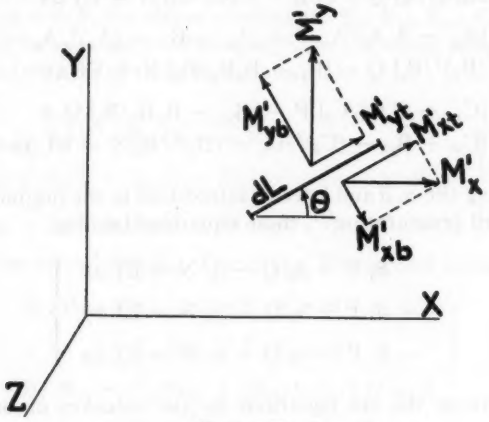


FIGURE V.

$$\left.\begin{aligned} M'_{xt} &= M'_x \cos\theta \\ M'_{xb} &= M'_x \sin\theta \end{aligned}\right\} \tag{37.}$$

$$\left.\begin{aligned} M'_{yt} &= M'_y \sin\theta \\ M'_{yb} &= M'_y \cos\theta \end{aligned}\right\} \tag{38.}$$

And summing up the components of torsion and bending,

$$\left.\begin{aligned} M'_t &= M'_{xt} + M'_{yt} = M'_x \cos\theta + M'_y \sin\theta \\ M'_b &= M'_{xb} - M'_{yb} = M'_x \sin\theta - M'_y \cos\theta \end{aligned}\right\} \tag{39.}$$

Since M'_t, M'_b and M'_z are mutually perpendicular, then equation (8) would be written,

$$W = \frac{1}{2\,EI}\int\Big[1.3\,(M'_x\cos\theta + M'_y\sin\theta)^2 + (M'_x\sin\theta - M'_y\cos\theta)^2 + k_z\,M'^2_z\Big]\,dL \tag{40.}$$

Expanding and collecting terms, the equation may be rewritten,

$$W = \frac{1}{2\,EI}\int\Big[(1.3\cos^2\theta + \sin^2\theta)\,M'^2_x + (1.3\sin^2\theta + \cos^2\theta)\,M'^2_y + 0.6\sin\theta\cos\theta\,M'_x M'_y + k_z M'^2_z\Big]dL \qquad (41.)$$

The terms due to $0.6\sin\theta\cos\theta\,M'_x M'_y$ have been omitted in the previous-published section forms[1],[2] as giving rise to certain "secondary" effects. In order to check these section forms, this term may therefore be neglected, giving rise to the energy equation,

$$W = \frac{1}{2\,EI}\int\Big[(1.3\cos^2\theta + \sin^2\theta)\,M'^2_x + (1.3\sin^2\theta + \cos^2\theta)\,M'^2_y + k_z M'^2_z\Big]dL \qquad (42.)$$

The k's of equation (8) may then be defined as follows:

$k_x = (1.3\cos^2\theta + \sin^2\theta) = 1 + 0.3\cos^2\theta$

$k_y = (1.3\sin^2\theta + \cos^2\theta) = 1 + 0.3\sin^2\theta$

$k_z = 1$ for straight pipe

$=$ Von Karman flexibility factor for curved pipe.

8. Introduction to Symbolic Equations.

In handling the general equations for the case of involved piping arrangements, the symbolism so far used becomes quite cumbersome. To reduce the complexity of the resulting equations it has been found advisable to introduce a mathematical symbolism. By means of this symbolism the required equations can be simply and compactly written for the preliminary work. Of course, such symbolism cannot be carried through for the actual solution of the equations—and at that stage must be expanded out to the original equations.

This method has been built up on an external set of forces and moments acting at each of the anchors, and always referred to the origin, i. e. replacing the set at the anchor by an equivalent set at the origin. Physically, for an anchor that is not at the origin, this is equivalent to connecting the anchor by means of

an inflexible inextensible coupling to the origin. The forces and moments then act on this coupling, although, actually they are transmitted through to the anchor. On being transmitted through to the anchor, the forces remain unchanged—but the moments, of course, will be altered by such translation.

In starting a complex piping system, each anchor or restraint (partial anchor) is assigned a number, 1, 2, 3,------. Let "α" represent a system of forces and moments, and the anchor that it acts through be denoted by a subscript. Therefore, in general α_k represents the external forces and moments, referred to the origin, that act on the k-th anchor.

A symbolism must be introduced for the coefficients, A's, B's and C's—the results of the integrations. Thus let (k) represent the coefficients for the sections of pipe on which α_k acts. Similarly let (k-n) represent the coefficients for the sections of pipe on which the sum of $\alpha_k + \alpha_n$ acts. And (k-n) represents the coefficients for the sections of pipe on which the sum $(\alpha_k + \alpha_l + \alpha_m + \alpha_n)$ acts on. Also, in the new presentation, the equations of equilibrium may be restated at this point, namely, that $\sum_{i=1}^{n} \alpha_i = 0$ for a pipe with "n" anchors.

Now a point of junction of several branches has the same absolute rotation and movement through whichever branch it is approached. Thus, in Figure VI,

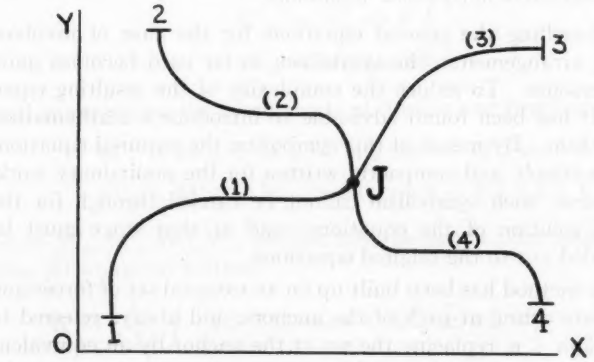


FIGURE VI.

$$(1)\ \alpha_1 = -EI\,(\phi_J - \phi_1)$$

$$(2)\ \alpha_2 = -EI\,(\phi_J - \phi_2)$$

Therefore,

$$(1)\ \alpha_1 - (2)\ \alpha_2 = -EI(\phi_2 - \phi_1)$$

And,

$$(1)\ \alpha_1 = EI\,[\tau\,(x_J - x_1) - (m_{xJ} - m_{x1}) - (\bar{y}_J\,\phi_J - \bar{y}_1\,\phi_1)]$$

$$(2)\ \alpha_2 = EI\,[\tau\,(x_J - x_2) - (m_{xJ} - m_{x2}) - (\bar{y}_J\,\phi_J - \bar{y}_2\,\phi_2)]$$

Therefore,

$$\begin{aligned}(1)\ \alpha_1 - (2)\ \alpha_2 &= EI\,[\tau\,(x_2 - x_1) - (m_{x2} - m_{x1}) \\ &\qquad - (\bar{y}_2\,\phi_2 - \bar{y}_1\,\phi_1)] \\ &= EI\,[\Delta x_{1-2} - \delta x_{1-2} - \Delta\,(\bar{y}\phi)_{1-2}]\end{aligned}$$

The reversal in sign for the term (2) α_2 is due to the fact that the integration for the coefficients proceeds from 1 to J, and from 2 to J instead of from J to 2. This fact is inherent in setting up the equations, namely, that the integration proceeds from the point of attack of the α's, the anchor, into the pipe.

To be more specific on the symbolism, and what the symbolic equation means, consider the following symbolic equations for which the expanded equivalents are given in terms of the notation used in equations (30):

a.) Rotation equation.

$$\begin{aligned}-EI\,\Delta\phi_y &= (4)\,\alpha_4 \\ &= ({}_4C_s)\,{}_4M_{0y} - ({}_4C_z)\,P_4 + ({}_4C_x)\,N_4\end{aligned}$$

b.) Deflection equation.

$$\begin{aligned}EI\,[\Delta x - \delta x - \Delta\,(\bar{y}\phi_z) + \Delta\,(\bar{z}\phi_y)] &= (5)\,\alpha_5 \\ &= ({}_5A_y)\,{}_5M_{0z} - ({}_5C_z)\,{}_5M_{0y} + ({}_5A_{y^2} + {}_5C_{z^2}^{\,2})\,P_5 \\ &\qquad - ({}_5A_{xy})\,Q_5 - ({}_5C_{xz})\,N_5\end{aligned}$$

c.) Rotation Equation.

$$
\begin{aligned}
&- EI\,\Delta\phi_y \\
&= (1)\,\alpha_1 + (1 \rightarrow 2)\,(\alpha_1 + \alpha_2) + (1 \rightarrow 3)\,(\alpha_1 + \alpha_2 + \alpha_3) \\
&= [(1) + (1 \rightarrow 2) + (1 \rightarrow 3)]\,\alpha_1 + [(1 \rightarrow 2) \\
&\qquad + (1 \rightarrow 3)]\,\alpha_2 + (1 \rightarrow 3)\,\alpha_3 \\
&= [{}_1C_s + {}_{1\rightarrow 2}C_s + {}_{1\rightarrow 3}C_s]\,{}_1M_{0y} - [{}_1C_z + {}_{1\rightarrow 2}C_z \\
&\qquad + {}_{1\rightarrow 3}C_z]\,P_1 + [{}_1C_x + {}_{1\rightarrow 2}C_x + {}_{1\rightarrow 3}C_x]\,N_1 \\
&\qquad + [{}_{1\rightarrow 2}C_s + {}_{1\rightarrow 3}C_s]\,{}_2M_{0y} - [{}_{1\rightarrow 2}C_z \\
&\qquad + {}_{1\rightarrow 3}C_z]\,P_2 + [{}_{1\rightarrow 2}C_x + {}_{1\rightarrow 3}C_x]\,N_2 \\
&\qquad + [{}_{1\rightarrow 3}C_s]\,{}_3M_{0y} - [{}_{1\rightarrow 3}C_z]\,P_3 + [{}_{1\rightarrow 3}C_x]\,N_3.
\end{aligned}
$$

These should suffice to show how to handle the symbolic equations. It is to be noted that the symbolic terms are types of operators and will have different values in different equations; e. g. α_1 represents ${}_1M_{0z}$, P_1 and Q_1 in the equation for $\Delta\phi_z$, and it represents ${}_1M_{0y}$, P_1 and N_1 in the equation for $\Delta\phi_y$, etc. Similarly, (1) represents the terms ${}_1A_s$, ${}_1A_y$ and ${}_1A_x$ in the $\Delta\phi_z$ equation, and it represents ${}_1A_y$, ${}_1C_z$, $({}_1A_{y^2} + {}_1C_{z^2})$, ${}_1C_{zx}$ and ${}_1A_{xy}$ in the Δx equation, etc.

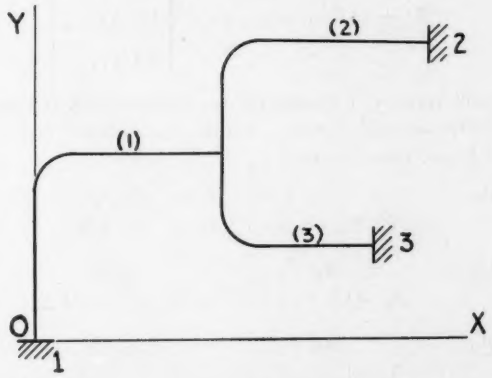
An attempt will be made to cover some of the piping problems that may arise. The types of anchors that we may have in practice will have to be described first. The most important type of anchor is of course the rigid anchor. This type of anchor consists of six degrees of restraint, 3 moment restraints and 3 force restraints. A summary of various types of anchors, of varying degree of restraint, is given in Table I.

TABLE I.

Anchor—Type and Description.	*No. of Restraints. Forces.*	*Moments.*
Rigid	3	3
Link parallel to an axis	1	0
Two links parallel to two axes Roller with freedom along an axis Link parallel only to one plane (at angle with axes of that plane)	2	0
Hinged anchor Link at angle with all axes	3	0
Guided sleeve parallel to an axis	2	2
Guided sleeve parallel to an axis, ξ guided against torsion	2	3

In general, for a single plane system, the number of equations necessary to solve simultaneously will be the total number of external restraints minus three, and for the space problem will be the total number of external restraints minus six. The remaining unknowns are to be found from conditions of equilibrium, $\Sigma\alpha = 0$.

Case I. The Three-Anchor Single Plane Problem.



If all anchors are assumed rigid, then, between anchors 1 and 2, and between anchors 1 and 3, these equations may be written: namely,

$$\left.\begin{aligned}(1)\,\alpha_1 - (2)\,\alpha_2 &= \begin{cases} -\,EI\,\Delta\phi_{1-2} = 0 \\ EI\,\Delta x_{1-2} \\ EI\Delta\, y_{1-2} \end{cases} \\ (1)\,\alpha_1 - (3)\,\alpha_3 &= \begin{cases} -\,EI\,\Delta\phi_{1-3} = 0 \\ EI\,\Delta x_{1-3} \\ EI\,\Delta y_{1-3} \end{cases}\end{aligned}\right\} \qquad (a)$$

These represent six equations, symbolically written, but as written they contain nine unknowns: three each for α_1, α_2 and α_3. However,

$$\alpha_1 + \alpha_2 + \alpha_3 = 0 \qquad (b)$$

Therefore, using the relation (b), equations (a) can be rewritten as

$$\left.\begin{aligned} (1)\,\alpha_1 \qquad\qquad - (2)\,\alpha_2 &= \begin{cases} 0 \\ \text{EI}\,\Delta x_{1-2} \\ \text{EI}\,\Delta y_{1-2} \end{cases} \\ [(1)+(3)]\,\alpha_1 + (3)\,\alpha_2 &= \begin{cases} 0 \\ \text{EI}\,\Delta x_{1-3} \\ \text{EI}\,\Delta y_{1-3} \end{cases} \end{aligned}\right\} \qquad (c)$$

These now represent six equations, symbolically rewritten, and contain only six unknowns. From the general equations, in expanded form, these become:

$$\left.\begin{aligned} {}_1A_s\,{}_1M_0 \quad + {}_1A_y\,P_1 \quad - {}_1A_x\,Q_1 \\ - {}_2A_s\,{}_2M_0 - {}_2A_y\,P_2 + {}_2A_x\,Q_2 &= 0 \\ {}_1A_y\,{}_1M_0 \quad + {}_1A_{y^2}\,P_1 \quad - {}_1A_{xy}\,Q_1 \\ - {}_2A_y\,{}_2M_0 - {}_2A_{y^2}\,P_2 + {}_2A_{xy}\,Q_2 &= \text{EI}\,\Delta x_{1-2} \\ - {}_1A_x\,{}_1M_0 \quad - {}_1A_{xy}\,P_1 \quad + {}_1A_{x^2}\,Q_1 \\ + {}_2A_x\,{}_2M_0 + {}_2A_{xy}\,P_2 - {}_2A_{x^2}\,Q_2 &= \text{EI}\,\Delta y_{1-2} \\ ({}_1A_s + {}_3A_s)\,{}_1M_0 + ({}_1A_y + {}_3A_y)\,P_1 - ({}_1A_x + {}_3A_x)\,Q_1 \\ + {}_3A_s\,{}_2M_0 + {}_3A_y\,P_2 - {}_3A_x\,Q_2 &= 0 \\ ({}_1A_y + {}_3A_y)\,{}_1M_0 + ({}_1A_{y^2} + {}_3A_{y^2})\,P_1 - ({}_1A_{xy} + {}_3A_{xy})\,Q_1 \\ + {}_3A_y\,{}_2M_0 + {}_3A_{y^2}\,P_2 - {}_3A_{xy}\,Q_2 &= \text{EI}\,\Delta x_{1-3} \\ -({}_1A_x + {}_3A_x)\,{}_1M_0 - ({}_1A_{xy} + {}_3A_{xy})\,P_1 + ({}_1A_{x^2} + {}_3A_{x^2})\,Q_1 \\ - {}_3A_x\,{}_2M_0 - {}_3A_{xy}\,P_2 + {}_3A_{x^2}\,Q_2 &= \text{EI}\,\Delta y_{1-3} \end{aligned}\right\} \qquad (d.)$$

Where all numerical subscripts refer to the branch designated by the subscript.

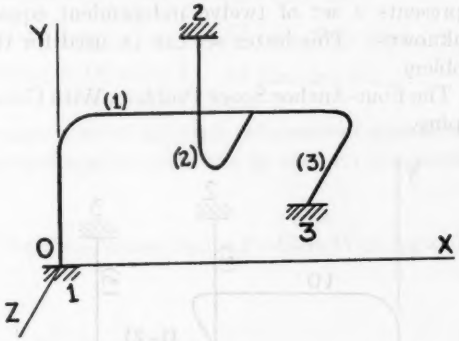
A set of equations between anchors 2 and 3 could have been written,

$$(2)\,\alpha_2 - (3)\,\alpha_3 \pm \begin{cases} -\text{EI}\,\Delta\phi_{2-3} = 0 \\ \text{EI}\,\Delta x_{2-3} \\ \text{EI}\,\Delta y_{2-3} \end{cases} \qquad (e)$$

Offhand it might seem that equations (a) and (e), comprising a set of nine equations containing nine unknowns, could be solved

for the unknowns. However, the equations (e) are not independent of equations (a), as both branches (2) and (3) were already used in writing equations (a), and, therefore, they could not be solved simultaneously. Thus, equations (e) could have been arrived at by subtracting the first from the second, of equations (a). It is to be noted, however, that any two of the three sets of equations (a) and (e) could have been used with relation (b) to arrive at a solution.

Case II. The Three-Anchor Space Problem.



If all anchors are assumed rigid, then,

$$(1)\ \alpha_1 - (2)\ \alpha_2 = \left\{\begin{array}{l} -\ EI\ \Delta\phi_{x_{1-2}} = 0 \\ -\ EI\ \Delta\phi_{y_{1-2}} = 0 \\ -\ EI\ \Delta\phi_{z_{1-2}} = 0 \\ EI\ \Delta x_{1-2} \\ EI\ \Delta y_{1-2} \\ EI\ \Delta z_{1-2} \end{array}\right\} \qquad (a)$$

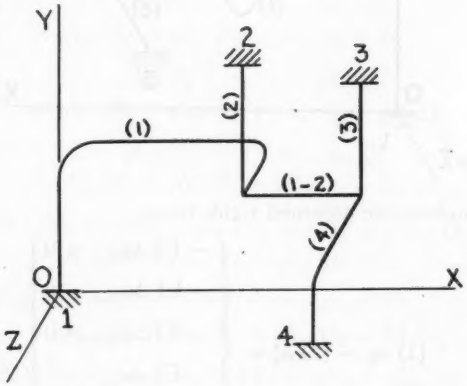
$$(1)\ \alpha_1 - (3)\ \alpha_3 = \left\{\begin{array}{l} -\ EI\ \Delta\phi_{x_{1-3}} = 0 \\ -\ EI\ \Delta\phi_{y_{1-3}} = 0 \\ -\ EI\ \Delta\phi_{z_{1-3}} = 0 \\ EI\ \Delta x_{1-3} \\ EI\ \Delta y_{1-3} \\ EI\ \Delta z_{1-3} \end{array}\right\}$$

This represents twelve equations with eighteen unknowns. Using $\alpha_1 + \alpha_2 + \alpha_3 = 0$,

$$\left.\begin{aligned} (1)\,\alpha_1 - (2)\,\alpha_2 &= \begin{cases} -\,EI\,[\Delta\phi_x, \Delta\phi_y, \Delta\phi_z]_{1\text{-}2} = 0 \\ EI[\,\Delta x, \Delta y, \Delta z]_{1\text{-}2} \end{cases} \\ [(1) + (3)]\,\alpha_1 + (3)\,\alpha_2 &= \begin{cases} -\,EI\,[\Delta\phi_x, \Delta\phi_y, \Delta\phi_z]_{1\text{-}3} = 0 \\ EI\,[\Delta x, \Delta y, \Delta z]_{1\text{-}3} \end{cases} \end{aligned}\right\} \quad (b)$$

which represents a set of twelve independent equations with twelve unknowns. This latter set can be used for the solution of the problem.

Case III. The Four-Anchor Space Problem, With Common Junction Piping.



If all anchors are assumed rigid, then,

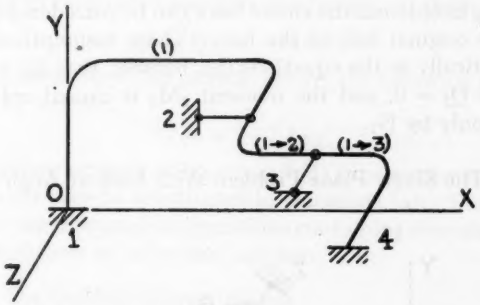
$$\left.\begin{aligned} (1)\,\alpha_1 - (2)\,\alpha_2 &= \begin{cases} -\,EI\,[\Delta\phi_x, \Delta\phi_y, \Delta\phi_z]_{1\text{-}2} = 0 \\ EI[\Delta x, \Delta y, \Delta z]_{1\text{-}2} \end{cases} \\ (1)\,\alpha_1 + (1\text{-}2)\,(\alpha_1 + \alpha_2) - (3)\,\alpha_3 &= \begin{cases} -\,EI\,[\Delta\phi_x, \Delta\phi_y, \Delta\phi_z]_{1\text{-}3} = 0 \\ EI\,[\Delta x, \Delta y, \Delta z]_{1\text{-}3} \end{cases} \\ (1)\,\alpha_1 + (1\text{-}2)\,(\alpha_1 + \alpha_2) - (4)\,\alpha_4 &= \begin{cases} -\,EI\,[\Delta\phi_x, \Delta\phi_y, \Delta\phi_z]_{1\text{-}4} = 0 \\ EI\,[\Delta x, \Delta y, \Delta z]_{1\text{-}4} \end{cases} \end{aligned}\right\} \quad (a)$$

And since $\alpha_1 + \alpha_2 + \alpha_3 + \alpha_4 = 0$, combining terms of common unknowns, the above set may be rewritten,

$$\left.\begin{aligned}
(1)\,\alpha_1 - (2)\,\alpha_2 &= \begin{cases} -\,EI\,[\Delta\phi_x, \Delta\phi_y, \Delta\phi_z]_{1-3} = 0 \\ EI\,[\Delta x, \Delta y, \Delta z]_{1-3} \end{cases} \\
[(1) + (1\text{-}2)]\,\alpha_1 + (1\text{-}2)\,\alpha_2 & \\
\quad - (3)\,\alpha_3 &= \begin{cases} -\,EI\,[\Delta\phi_x, \Delta\phi_y, \Delta\phi_z]_{1-3} = 0 \\ EI\,[\Delta x, \Delta y, \Delta z]_{1-3} \end{cases} \\
[(1) + (1\text{-}2) + (4)]\,\alpha_1 + & \\
[(1\text{-}2) + (4)]\,\alpha_2 + (4)\,\alpha_3 &= \begin{cases} -\,EI\,[\Delta\phi_x, \Delta\phi_y, \Delta\phi_z]_{1-4} = 0 \\ EI\,[\Delta x, \Delta y, \Delta z]_{1-4} \end{cases}
\end{aligned}\right\} \quad (b)$$

This represents a set of eighteen independent equations containing eighteen unknowns which can be used for the solution of the problem.

Case IV. The Two-Anchor Space Problem With Links Parallel to Axes.



If anchors 1 and 4 are assumed rigid, then,

$$\left.\begin{aligned}
&(1)\,\alpha_1 + (1\rightarrow 2)\,(\alpha_1 + \alpha_2) \\
&+ (1\rightarrow 3)\,(\alpha_1 + \alpha_2 + \alpha_3) = \begin{cases} -\,EI\,[\Delta\phi_x, \Delta\phi_y, \Delta\phi_z]_{1-4} = 0 \\ EI\,[\Delta x, \Delta y, \Delta z]_{1-4} \end{cases} \\
&(1)\,\alpha_1 = EI\,[\Delta x_{1-2} - \bar{y}_2\,\phi_{z2} + \bar{z}_2\,\phi_{y2}] \\
&(1)\,\alpha_1 + (1\rightarrow 2)\,(\alpha_1 + \alpha_2) = EI\,[\Delta z_{1-3} - \bar{x}_3\phi_{y3} + \bar{y}_3\phi_{x3}]
\end{aligned}\right\} \quad (a)$$

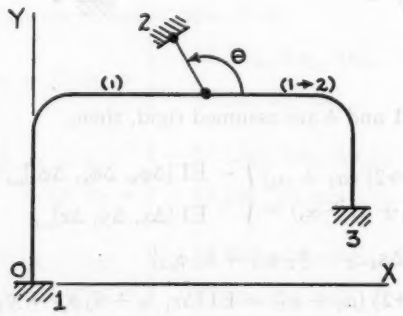
This represents eight equations, but as written contains twenty-two unknowns: six each for α_1, α_2 and α_3, and ϕ_{y2}, ϕ_{z2}, ϕ_{x3}, ϕ_{y3}. However, there are the following auxiliary equations, namely,

$$\left.\begin{aligned} &(1)\ \alpha_1 = -\,EI\,(\phi_{y2},\ \phi_{z2}) \\ &(1)\ \alpha_1 + (1\rightarrow 2)\ (\alpha_1 + \alpha_2) = -\,EI\,(\phi_{x3},\ \phi_{y3}) \\ &\text{Also,} \\ &{}_2M_{0x} = 0;\ {}_2M_{0y} = +\,\bar{z}_2\,P_2;\ {}_2M_{0z} = -\,\bar{y}_2\,P_2;\ Q_2 = N_2 = 0. \\ &\text{And,} \\ &{}_3M_{0x} = +\,\bar{y}_3\,N_3;\ {}_3M_{0y} = -\,\bar{x}_3\,N_3;\ {}_3M_{0z} = 0;\ P_3 = Q_3 = 0. \end{aligned}\right\} \quad (b)$$

The equations (b) represent fourteen auxiliary equations, which when substituted in equations (a) reduce that set to eight independent equations containing eight unknowns.

The assumption has been made in this case that the link applies a force only in its axial direction and cannot support a moment. Also, when the pipe expands, the change in angle of the link is so small that components of the force along the remaining two axes can be neglected, and the entire force can be considered as acting along the original axis of the link. These assumptions appear mathematically in the equations (b), namely, that $Q_2 = N_2 = 0$ and $P_3 = Q_3 = 0$, and the moment ${}_2M_0$ is caused only by P_2, and ${}_3M_0$ only by N_3.

Case V. The Single Plane Problem With Link at Angle to Axes.



If anchors 1 and 3 are assumed rigid, then the following equations may be written;

$$\left.\begin{aligned} (1)\ \alpha_1 + (1 \rightarrow 2)\ (\alpha_1 + \alpha_2) &= \begin{cases} -\ EI\ \Delta\phi_{1-3} = 0 \\ EI\ [\Delta x,\ \Delta y]_{1-3} \end{cases} \\ (1)\ \alpha_1 &= \begin{cases} EI\ [\Delta x_{1-2} - \delta x_2 - \bar{y}_2\ \phi_2] \\ EI\ [\Delta y_{1-2} - \delta y_2 + \bar{x}_2\ \phi_2] \end{cases} \end{aligned}\right\} \quad (a)$$

This represents five equations containing nine unknowns: three each for α_1 and α_2, and δx_2, δy_2 and ϕ_2.

Using the same assumptions for the link as was made in Case IV, the force along the link may be resolved into components parallel to the axes, so that

$$\left.\begin{aligned} Q_2 &= P_2 \tan\theta \\ \text{And, therefore,} \\ {}_2M_0 &= \bar{x}_2\, Q_2 - \bar{y}_2\, P_2 = P_2\ (\bar{x}_2 \tan\theta - \bar{y}_2) \end{aligned}\right\} \quad (b)$$

Thus α_2 can be expressed in terms of one unknown only, P_2. Also, the movement at the link can only take place perpendicular to the link, and since the change in angle has been neglected, the slope of the force along the link and the slope of the movement are inverse to each other and opposite in sign, so that

$$\delta x_2/\delta y_2 = -\tan\theta \quad (c)$$

From fundamentals of the method set up, the following equation may also be written,

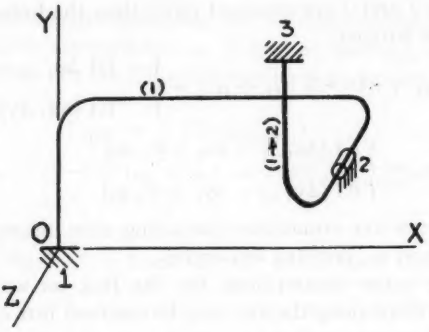
$$(1)\ \alpha_1 = -\ EI\ \phi_2. \quad (d)$$

The auxiliary equations (b), (c) and (d) express four additional relations that may be substituted in equations (a). This results in a set of five independent equations containing five unknowns, which is sufficient to solve this problem.

Case VI. The Guided Sleeve Problem.

If anchors 1 and 3 are assumed rigid, and a guided sleeve exists at 2, then the following symbolic equations may be written,

$$\left.\begin{aligned} (1)\ \alpha_1 &= \begin{cases} -\ EI[\Delta\phi_x,\ \Delta\phi_y]_{1-2} = 0 \\ EI\ [\Delta x,\ \Delta y]_{1-2} \end{cases} \\ (1)\ \alpha_1 + (1 \rightarrow 2)\ (\alpha_1 + \alpha_2) &= \begin{cases} -\ EI\ [\Delta\phi_x,\ \Delta\phi_y,\ \Delta\phi_z]_{1-3} = 0 \\ EI\ [\Delta x,\ \Delta y,\ \Delta z]_{1-3} \end{cases} \end{aligned}\right\} \quad (a)$$

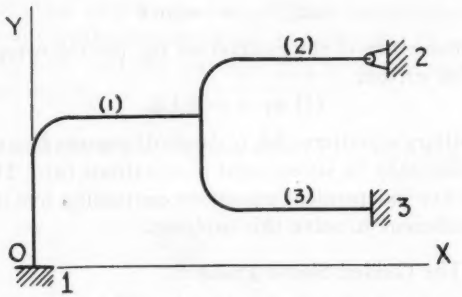


Now if we substitute the following relations in the above equations, namely,

$$\left.\begin{aligned} N_2 &= 0 \\ {}_2M_{0z} &= Q_2\,\bar{x}_2 - P_2\,\bar{y}_2 \end{aligned}\right\} \qquad (b)$$

there now results a set of ten independent equations containing ten unknowns.

Case VII. Single Plane Branch Problem With One Hinged Anchor.



If anchors 1 and 3 are rigid, and anchor 2 is hinged, then,

$$\left.\begin{aligned} (1)\,\alpha_1 - (2)\,\alpha_2 &= \left\{\begin{matrix} EI\,[\Delta x - \bar{y}_2\,\phi_2]_{1-3} \\ EI\,[\Delta y + \bar{x}_2\,\phi_2]_{1-3} \end{matrix}\right. \\ [(1)+(3)]\,\alpha_1 + (3)\,\alpha_2 &= \left\{\begin{matrix} -EI\Delta\,\phi_{1-3} = 0 \\ EI\,[\Delta x,\ \Delta y]_{1-3} \end{matrix}\right. \end{aligned}\right\} \qquad (a)$$

where use has already been made of the conditions of equilibrium; namely, $-\alpha_3 = \alpha_1 + \alpha_2$.

Since the moment at anchor 2 is zero, for the hinged anchor, then there exists the relation

$$_2M_0 = \bar{x}_2 Q_2 - \bar{y}_2 P_2 \qquad (b)$$

And from fundamentals,

$$(1)\ \alpha_1 - (2)\ \alpha_2 = -EI\ \Delta\phi_{1-2} = -EI\ \phi_2. \qquad (c)$$

Substituting these two auxiliary relations back into the first set of equations, there results a set of five equations containing five unknowns.

9. The Symmetrical Equations.

The solution of the simultaneous equations, particularly as the number of branches, and or restraints, increases beyond those of the simple two-anchor problem, becomes quite laborious and time-consuming.

One of the features of the equations used is that the coefficients of the unknowns may be arranged in determinant form so that they are symmetrical about the principal diagonal. The terms along the principal diagonal are all positive, and symmetrically placed terms on both sides of the diagonal are equal. This can be done by writing all the equations so that they always start from the same anchor and eliminating the set of attacking forces and moments at that common anchor. In the symbolism this means writing the equations, say, from 1 to 2, 1 to 3, 1 to 4, etc., and then letting $\alpha_1 = -\alpha_2 - \alpha_3 - \alpha_4$--------. When this is done the arrangement of equations and unknowns for this symmetry is readily observable. Several such arrangements will be possible.

The solution of such symmetrical equations lends itself to short-cut methods. Such methods will not be discussed here, but one is referred to The Kellogg Company's "Design of Piping Systems"[3] for a discussion of one such method.

10. The Solution for the Movements of a Point in the System.

Occasionally it becomes important to find the movements of a particular point in the piping system. This may occur, for example, where ship-board piping passes through a bulkhead and is "floated" through by means of a bellows. It then becomes

necessary to know the movements at the point of attaching the bellows so that these movements may be specified in ordering the bellows.

Such movements can be determined directly from the general equations once the numerical values for the unknowns are determined at each of the anchors. The rotation equations may be written from the nearest anchor to the point of interest, and using the coefficients of the pipe from the anchor to that point, and the numerical values of the forces and moments previously determined, the rotations at the point may be determined. Then using the deflection equations in the same manner, the only unknowns will now be δx, δy and δz, and they can be readily determined.

11. The Solution for Vari-Sized Piping and Change of Modulus.

When the piping in an arrangement varies in size, an assumption made in the energy equation (8) becomes invalid. The term "I" cannot then be removed from under the integral sign. This will result in terms for the coefficients, equations (29), of the type

$$\text{Coefficient} = \int \frac{k\, f(x, y, z)}{I}\, dL$$

Since the piping is usually uni-sized for any one particular branch, this would result in coefficients of the type

$$(1)/I_1;\ (2)/I_2\ ;(3)/I_3;\ \text{etc.}$$

The values of I will then disappear from the terms in the equations that are functions of ϕ, Δx, Δy, etc.

Another method of handling this situation is to use a basic size of piping for a basic value of I. Then the coefficients, in terms of the I's will appear as

$$(1);\ (2)I_1/I_2;\ (3)\ I_1/I_3;\ \text{etc.},$$

where I_1 is chosen as the basic I. The terms in the general equations that contain I will then be given in terms of the basic I, I_1 in this case.

In case "E" is not constant, the situation may be handled in the same manner. Such a case may arise where a branch of a system is at a different temperature than the remainder of the system. The material used is usually uniform so that E usually does not vary from this source.

12. Thrust Lines.

A thrust line, for a run of piping, may be defined as the line of action of the single force that equivalently replaces the system of forces and moments acting on that run of piping. By its eccentricity with respect to points on the pipe, that force will create the actual resultant moment at these points. From its very definition it is a line of "zero moment," or a "neutral axis." The slope and sense of the thrust line is that of the resultant force. Each component of a pipe system, (1), (2), (1-2), or (1→3), etc., acted on by its own set of forces and moments, α_1, α_2, $(\alpha_1 + \alpha_2)$, or $(\alpha_1 + \alpha_2 + \alpha_3)$, etc., respectively, will have its own thrust line.

The concept of thrust lines is of advantage in locating points of maximum moment. The thrust line also proves advantageous in choosing points of low moment for placing flanges in designing a piping system. Such design will decrease some of the flange troubles that occur in installations.

In general, the thrust line lies in space and may not pass through a point on the pipe. In the case of single plane piping, the thrust line lies in the plane of the piping, and its use permits the point of maximum stress to be found directly by observation. For the space problem, its use is more limited, and it is best handled by projecting both the piping and the thrust lines on each of the coordinate planes.

If a piping system be drawn to scale, such thrust lines may be added to the drawing. This may be done in two ways. The first method, though more laborious, is explained because it is believed to give a more physical picture of the concept of thrust lines. If the numerical solution has been obtained, then the forces and moments are known, and it becomes possible to obtain the moments at any points in the pipe. In general, if its plane is determined, a line is specified by two conditions. Let these two conditions be the moments at any two points, say the anchor and junction point. The thrust creating the moment in a given plane is the resultant of the forces acting in that plane. Dividing these moments by their resultant thrusts gives the moment arms at these points. If circles are now constructed, with radius equal to the moment arms, about these points, the thrust line is then tangent to these two circles. Account must be taken of the sense of the moments in choosing the proper tangent line.

The alternate, but simpler, method follows: Since the thrust line is a line of zero moment, then, for the projections on each plane, the equations of such lines will be,

X-Y plane. $M_{0z} = Qx - Py$

Y-Z plane. $M_{0x} = Ny - Qz$

Z-X plane. $M_{0y} = Pz - Nx$

It then becomes a simple matter to find any two points that satisfy each of the above equations, and the line through these points will be the thrust line.

13. Acknowledgments.

In the preparation of this paper acknowledgment is made of a liberal use of The Kellogg Company's "Design of Piping Systems"[3]. The actual use of the methods herein presented are based on the use of the "Meyer-Hovgaard"[1] or "Bleick"[2] section forms as a means of numerical integration.

The helpful suggestions and contributions to this development of Messrs. G. Adler, B. Garfinkel, H. and L. Miller and S. Zeitzoff of the calculating group at the New York Navy Yard is acknowledged. The author is particularly appreciative to Mr. Zeitzoff for his careful checking and proof-reading of the manuscript, and his timely suggestions and cooperation.

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SHIP MATERIAL SCHEDULING.

By Herbert M. Neuhaus.

Ship engineering periodicals and similar publications of the shipbuilding profession have frequently displayed articles describing methods of ship construction and the methods or means whereby the proper materials arrive at the shipways at the correct time to insure orderly erection. The majority of these articles have described but one phase of the erection: namely, hull erection. Occasionally the ship's machinery plant has been given passing mention, but rarely has the hull with its machinery been considered as an integral unit.

A review of ship assembly methods employed at various shipyards reveals three generally accepted shipbuilding methods. These are the "custom-built" job, the unit or sectional, and the continuous assembly. A short explanation of each will serve to illustrate the methods: In custom-built ships, each part is tailored to its adjacent assembly on the ways piece by piece. Careful fitting and consequent slow progress are involved. Much of the preliminary fitting is carried out in the shops through the use of molds and templates produced from full scale drawings on the mold loft floor. Obviously, the custom-built ship does not easily lend itself to quantity production. However, the availability of molds, templates, and patterns permits the reproduction of additional similar elements when the "bugs" in the first unit have been corrected and overcome.

To speed construction, the shipyard, if it has available sufficient ground space and crane capacity, may build or assemble the small parts into larger units or subassemblies before erection into the final assembly. These subassemblies may be increasingly larger as the space for their erection is enlarged and greater crane capacity is made available for handling the heavier weights. However, conventional methods are still followed in erection. Most of the gain is due to duplication. In the custom-built ship, only one or two, rarely more, vessels are built simultaneously, particularly when no identical ships are to follow. Where several

vessels of the same design follow rapidly in succession, work is put through the shops in batches of from two to six or more ships. Instead of forming one plate to the necessary shape, six will be cut and formed at one set-up. Large units also permit the ready assembly of piping and cable into the sections, only connecting up being required in final assembly. Cheaper production methods follow as more efficient methods of cutting, forming, assembly and erection become apparent with repetition. Labor itself becomes more efficient, though not more highly skilled, as workmen are confined to single or few tasks at which they become adept due to repetitive performance.

Using the feature of repetitive performance, the shipbuilder, if constructing even medium sized vessels, can use progressive assembly methods. Starting in his shops, material takes shape and passes in its semi-fabricated form to an assembly line where the various units follow one another in an ordered sequence into the completed vessel. Whether the vessel is built upside down and later rolled over, laid on a movable carriage and transferred from one station to another on the assembly line, or assembled progressively on a series of ways makes but little difference as the erection order of hull and machinery parts is the same. Again the workmen are assigned repetitive tasks at which they reach a definite efficiency without too much preliminary training.

Further study of the picture reveals two other factors which govern directly the rate of erection of a vessel and to some extent, the order of erection. These two are the availability of ways and the location of large crane handling facilities. The availability of ways governs the start of major erection by definitely establishing from the standpoint of time, the dates of keel laying and launching. Launching is determined by the need for ways and the watertightness, strength, and stability of the hull insofar as its ability to be waterborne is concerned. Somewhere near the launching date shipyard crane capacity is a major feature affecting erection. If the ways crane capacity is greater than that of the fitting-out piers, heavy units such as boilers are installed before launching. If the fitting-out pier crane capacity is greater, then heavy units are installed after launching. In the former, time on the ways is longer than when components are placed after launching. Some shipyards are fortunate to have suitable capacity at either location. The guiding factor then becomes availability and capacity of the ways. One other condition may

affect installation arrangements. This is the availability of drydocks or marine railways. Where no such facilities are available, the shipbuilder may install underwater features such as stern tube shafting and propellers before launching. In a few cases some builders use divers to accomplish the same result while others use nearby drydocking facilities just prior to delivery of a vessel to its purchaser.

Observation of shipbuilding procedure over a period of years reveals that for a given design of vessel building at several shipyards, a very close symmetry exists in the erection order of hull units and components. This same sequence is found in operations of the drafting room, the mold loft, the various shops, and the ways. In most cases the design is carried forward from a system of ship structure group planning which starts with one section of the ship and moves forward or aft and upward as individual sections are completed. The drafting room precedes the mold loft, the mold loft the shops, and the shops the ways. Meanwhile the production departments and purchasing departments endeavor to keep ahead of the shops in placing orders and obtaining material deliveries.

In 1935 preliminary studies of ship erection procedures were undertaken. Data taken from the actual accomplished dates on then completed modern destroyers, cruisers, and submarines revealed that despite wide geographical disposition of shipyards, either Goverment or privately operated, and differing dates of construction, a definite pattern of ship erection was evident for the same types of vessels. The assembly of parts or components into the growing ships was dependent upon the rate of erection of the hulls. Subsequent studies indicated that the method of hull assembly caused little or no change in a vessel's erection order or the priority of assembly and for practical purposes that feature could be neglected. Shortly before the preliminary amalgamation of the Bureau of Construction and Repair and the Bureau of Engineering, Germany invaded Poland touching off the present second World War. The Navy Department scrapped its then existing ship construction plans and laid out an entirely new and enlarged plan as the evidence indicated that this country would require additional ship facilities. Ship construction was rushed to protect our interests on the seas.

The Congress of the United States granted a portion of the Navy's requests early in 1940, following within a few months

with an additional increase to give this country a Two Fleet Navy. With the granting of these authorizations and their accompanying appropriations, shipbuilding began with determination. Old yards were rehabilitated, existing shipyards were modernized and expanded, not alone in manufacturing facilities but also in their related production employee training and procurement activities. Competition by individual shipyards for the products of shipyard suppliers was in full force and controls were now required to assure equitable distribution of ship components. Some control was afforded by the preference rating system familiarly known as "priorities" applied to all products by the Office of Production Management. The system applied certain preferential treatments as a matter of quasi-law to all materials destined for particular uses. Civilian uses were considerably suppressed or absolutely forbidden to the gain of the military forces. However, the system broke down, due primarily to the assignment of the highest preferred ratings to the majority of products. As a result the Army, Navy, Maritime Commission, and certain agencies still remained in competition, directly and through their chains of contractors. A partial cure was attempted by establishing intermediate higher preference ratings but the result was a mere escalation of the competing agencies, again to the highest levels. The next cure offered was the short-lived Production Requirements Plan followed in turn by the Controlled Materials Plan which, by permitting the competing agencies, including Industry divisions of the War Production Board, to individually assign allocated quantities of the three most important materials, steel, copper, and aluminum, to the end uses or products which the agencies considered most vital to their effort. The next step was to direct manufacturers and agencies to schedule their own products and components including related sub-components. This was done by General Scheduling Order M-293 which gave a rather absolute control over the output of certain very critical or hard-to-obtain parts.

The Bureau of Ships had long foreseen this latter step anticipating the contingency in 1941 by initiating its own scheduling system, developed from the situations discussed in the preceding paragraphs. Ships had to be built in an orderly manner and destructive competition for materials eliminated, regulations or no regulations.

Scheduling of ship components actually begins when a ship-

builder receives his ship contract. In addition to the shipyard, the manufacturers of ship components are also interested. With several ships of the same design building at several yards, the same design component may be ordered by many shipyards from the same manufacturer at the same time, all calling for delivery within a short interval in the same calendar period. The impact upon a component manufacturer is tremendous, not only from the shop production angle but from the visits and calls from shipbuilders' expediters, purchasing agents, government inspection services, and a host of others. To relieve this deluge upon a manufacturer, a shipping date list indicating when material should be at the various shipyards was mandatory. This would enable a manufacturer to adjust his production schedule to its most efficient output and required that shipbuilders' needs be programmed on a standard all-inclusive basis.

One of the fundamental facts revealed by the previous mentioned study was—that practically all shipbuilders follow a similar order of erection of a vessel. A study of this nature, illustrated by Figure 1, covers the erection order of components in a typical small vessel. The various items are arranged in the sequence in which they follow each other into the ship during erection. Descriptions are general, as many include whole systems. The time or point of placement in the assembly marks the start of erection in the main hull assembly. For example, the main steam piping cannot be installed until pre-fabricated pipe sections, pipe fittings, flanges, bulkhead connections, valves, strainers, hangers, anchors, joints, and coupling bolts are available simultaneously. Items such as reduction gears, which though requiring careful installation, need only shims or chocks, holding-down bolts, and dowel pins to install, details which require little fabrication time.

Opposite each component item is listed a percentage figure representing the point of the ship's building period at which the item should be ready to land into the hull. For convenience, two columns of percentages are tabulated, one showing percentage before completion and one showing percentage from quasi-contract. This latter term requires explanation as it is one of the features which makes component scheduling possible. Each shipbuilder knows from his way schedules, or way occupancy charts, just when he can lay a keel and launch a vessel. If he has several identical ships following one another on the same

MATERIAL ERECTION SCHEDULE.

AUXILIARY VESSEL.

(Turbo-Reduction Drive.)

Pre-Keel Period—4 Months.

Erection Order Number.	Item.	Percentage of Quasi-Building Period.	Percentage Before Completion.
1.	Fuel Oil Transfer and Drainage Piping	30	70
2.	Stuffing Tubes	32	68
3	Underwater Sound—Hull Casting	35	65
4	Shaft Struts, Stern Tubes, Rudder Weldment	37	63
5.	Sea Chests	40	60
6.	Doors, Hatches, Manholes and Coamings	43	57
7.	Cable—All Systems	44	56
8.	Shafting and Bearings	46	54
9.	Rudder and Rudder Forgings	46	54
10.	Blowers, Forced Draft	47	53
11.	Main Boilers	48	52
12.	Hawse Pipes and Chain Pipes	49	51
13.	Lighting and Wiring Fixtures	49	51
14.	Surge Tanks (Deaerator and F. W. Heater)	50	50
15.	Plumbing Fixtures	50	50
16.	Sprinkling System—Magazines	50	50
17.	Condensers, Auxiliary	50	50
18.	Fuel Oil System Piping	50	50
19.	Lubricating Oil Piping	50	50
20.	Deck Drains and Scuppers	51	49
21.	Inclined Ladders	51	49
22.	Bitts and Chocks	51	49
23.	Uptakes—Below Deck	51	49
24.	H. P. and L. P. Air Piping	52	48
25.	Pumps, Fuel Oil	52	48
26.	Hull Insulation	52	48
27.	Fire and Flushing System Pumps	52	48
28.	Distribution Panels	52	48
29.	Feed Water Piping	53	47
30.	Salt Water Piping	53	47
31.	Airports	54	46
32.	Main and Auxiliary Steam Piping (Sketch or Pre-Fabricated)	54	46
33.	Distilling Plant (Evaporators)	54	46
34.	Emergency Diesel-Generator and Switchboard	54	46
35.	Refrigerating Plant	55	45
36.	Hoists and Winches—Ammunition	55	45
37.	Transformers	55	45
38.	H. P. Air Compressors	56	44

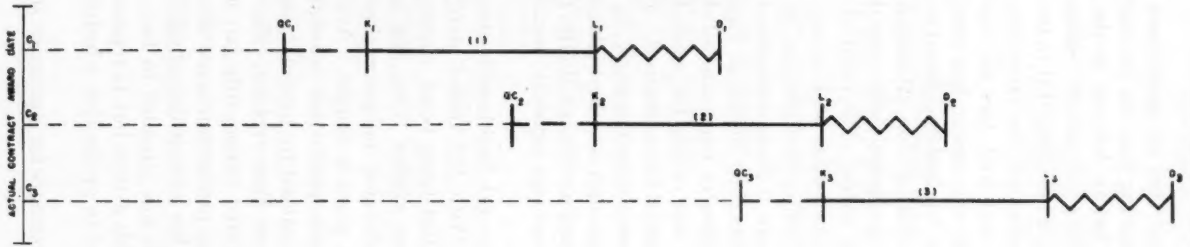
FIGURE 1.

ways, the way occupancy chart has an appearance similar to Figure 2, the solid line representing time on the ways, and the irregular line time overboard before delivery to the purchaser, or as commonly known, the fitting-out period. Should construction of each of these vessels have been awarded to the builder by the same contract or at approximately the same date, obviously the actual building period of each hull does not start with the date of contract award. If the ship design has been completed and all interferences eliminated, the building period of each vessel is approximately the same, making due allowance for speed gained by repetitive work, familiarity with the design, and increased efficiency of working forces. The point at which hull fabrication is begun is some time before keel laying, to provide sufficient local material that continuous erection may proceed once keels are laid. Furthermore, it is not conducive to economy or efficiency to start work on the three vessels of Figure 2 at the same time as the resulting inflow of materials will heavily tax storage facilities of the yard and crowd the work load into a peak so that a uniform level cannot be maintained. The number of vessels that can be processed simultaneously is, of course, dependent upon several factors such as: the builder's financial credit which affects his hiring and purchasing ability, the number of ways available, his process storage capacity, local labor supply, and others.

At the present time a steady work load level is almost mandatory for most shipbuilders. With 125 trades involved in the building of a man-of-war, skilled labor being scarce, and competitors for the available labor market being any nearby war goods manufacturer, any stabilizing influence that will hold workmen at the shipbuilding plant is sought. Availability of material resulting from uniform production scheduling is one answer. Violent labor surges caused by irregular material flow with its intermittent lay-offs are thus avoided. Slightly higher production costs may be incurred temporarily but in the final analysis costs are reduced since production is not slowed by the absence and sometimes loss of key men or the substitution of less skilled help. The foregoing is not intended to be a digression from the central theme of this article but is representative of the problems that are involved in production scheduling of ship construction.

To schedule parts and components for successive ships on the

CONSTRUCTION COMPARISONS



LEGEND

C_1, C_2, C_3 — CONTRACT DATE OF VESSEL
QC_1, QC_2, QC_3 — QUASI-CONTRACT DATE
K_1, K_2, K_3 — DATE OF KEEL LAYING
L_1, L_2, L_3 — DATE OF LAUNCHING
D_1, D_2, D_3 — DATE OF COMPLETION OR DELIVERY

SOLID HEAVY LINE ——— VESSEL ON WAYS
WAVY HEAVY LINE ∿∿∿ VESSEL LAUNCHED, NOT COMPLETED

FIGURE 2

same way requires then, that some starting point be established when scheduling should begin for individual vessels. As this point is at some time interval before keel laying it can be fixed by several methods. Only a few need be mentioned here. The point of origin may be selected as the time at which the first order must be placed with a sub-contractor to have the finished article arrive in time for installation. On certain vessels this may require placing orders for main reduction gears twelve to fourteen months before keel laying. During this period succeeding orders may be placed while the shipyard locally fabricates what it can until additional material is at hand for fabrication. Generally, the major difficulty at any shipyard, insofar as material is concerned, is not local fabrication of hull steel or wood framing and planking, but is the receipt of semi-completed or finished assemblies from sub-contractors or government sub-contractors.

Another curiously interesting fact revealed by the ship erection study was—that all vessels of the same type require approximately the same local fabrication time prior to keel laying. The difference in building periods of the same design vessel building at different yards lies principally in the assembly methds and experience available at those yards. Roughly, the pre-keel period used in the erection schedules consists of two parts, the first being the interval between the date orders are placed on the steel mills and the date steel is received by the shipyard. The second is the time difference between the receipt of plate and structural material at the yard and its subsequent fabrication to a state where erection on the ways can begin. The foregoing is open to challenge but is generally true. Principal of the opposing arguments is the shipbuilders' contention that as the keel to delivery interval is reduced the preliminary or "pre-keel" period must necessarily increase. This may or may not be true. Recent observations have indicated that the pre-keel period remains fairly constant and in many cases is reduced compatible with continued experience.

The discussions in the two preceding paragraphs establish a portion of the range through which the pre-keel period may be considered. The extreme is realized by discarding entirely the pre-keel period and stating as a precept that no true assembly begins until the keel is laid. However, the difficulty arises in that no recognition by way of earlier material deliveries is given the shipbuilder who reduces his keel-delivery interval when keel

laying marks the zero origin of construction. In practice recognition is given to this reduction, for actual fabrication and preliminary assemblies of components can be undertaken before keel laying has taken place instead of delaying until after the keel event.

A partial solution to the scheduling problem is now evident. Briefly, by establishing first a building period, and then placing against this building period in an ordered sequence the various components that are part of the ships construction, an erection schedule for a prototype vessel can be determined. The building period can be disposed of by summarizing what has been described into the statement "The building period is equal to the pre-keel period plus the keel-completion period." This building period will be termed the "quasi-building" period to distinguish it from the actual and the contract building periods. The apparent date of starting construction will be known as the "quasi-contract" date. Figure 2 also illustrates this convention at points QC_1, QC_2, and QC_3.

If the quasi-building periods for several vessels of the same prototype design are taken and have matched against them by dates the points at which the various components are ready to land in the vessel, conversion of the dates so established into a percentage of the quasi-building period will give a consecutive series of gauge points. By averaging and, if necessary, weighting, the gauge points for the several vessels, discarding those percentages which deviate seriously from the median after due investigation for the cause of the difference, an average or composite series is arrived at which represents an erection sequence or construction guide for a normal prototype ship. This is the erection schedule which can be used by any builder of that type vessel.

Figure 1 illustrated a typical erection schedule computed by the foregoing method. This particular schedule has also been reviewed and culled by the builders of that type vessel prior to issuance of the schedule, giving it the advantage of additional professional authority. The net effect has been to inject variations which may be the result of latest shipbuilding practices and represents a mutual solution for component procurement purposes. Items are arranged in the sequence of their installation. Some general items, as piping systems, represent the simultaneous availability of valves, strainers, pipe fittings, flanges, hangers,

bulkhead connections, and pipe sections, as these are placed on board the ship as a unit. Bulkhead stuffing tubes for electric cable are generally attached to bulkheads during their erection or pre-assembly. Many shipbuilders make a practice of installing such fittings while the bulkheads are still on the welding table or just before the bulkheads are ready to hoist into position.

Adjacent to the column captioned "Percentage of Quasi-Building Period" an additional column has been added to show the "Percentage Before Completion." Mathematically this is merely 100 per cent minus the percentage in the preceding column. More will be said of this later. Still another column shows the prefabrication time required to build each component prior to its landing in the vessel. For example, piping normally requires 60 to 90 days to cut, bend, form and apply flanges preliminary to its installation. In the upper left hand corner of the schedule is listed the pre-keel period in months which is to be used for computing the quasi-building period. For convenience and to reduce confusion the erection schedules normally issued to the field contain only the "Percentage Before Completion," the prefabrication time, and the pre-keel period, omitting the "Percentage of Quasi-Building Period."

With numerical factors established, computation of the actual required dates can be undertaken. The actual quasi-building period is equal to the keel to completion period expressed in months plus the proper pre-keel period expressed in months. If the keel laying date is September 15, 1943, and the completion date is January 1, 1945, there is an interval of 15½ months from keel to completion. Using a pre-keel period for this vessel of 4 months (see Figure 1), the quasi-building period will be 15½ + 4 = 19½ months. Selecting from the erection schedule the percentage for the component in question (Item 35, Refrigeration Plant, Figure 1), we find 55 per cent. The 55 per cent multiplied by 19½ gives an interval of 10.7 months from quasi-contract. The quasi-contract date must be obtained first and is found by subtracting the pre-keel period, 4 months, from the keel date, September 15, 1943, the result being May 15, 1943. Adding 10.7 months to May 15, 1943, gives a required date of April 6, 1944.

Occasionally, due to "tight" spots in the material availability picture or to very short assembly periods as in the case of small boats, percentage intervals are inconvenient due to the "compression" of dates. Here actual days or months before comple-

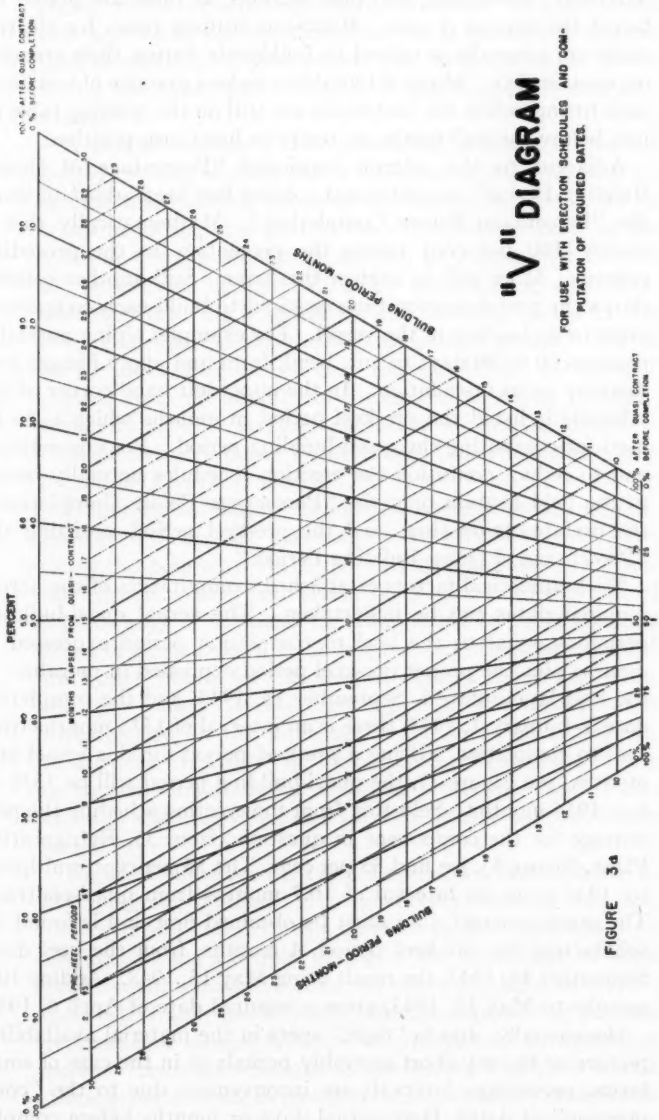

"V" DIAGRAM
FOR USE WITH ERECTION SCHEDULES AND COMPUTATION OF REQUIRED DATES.
PERCENT
MONTHS ELAPSED FROM QUAD - CONTRACT
PRE-KEEL PERIOD
BUILDING PERIOD - MONTHS
BUILDING PERIOD - MONTHS
PERCENT
100% AFTER QUAD CONTRACT
0% BEFORE COMPLETION

FIGURE 3a

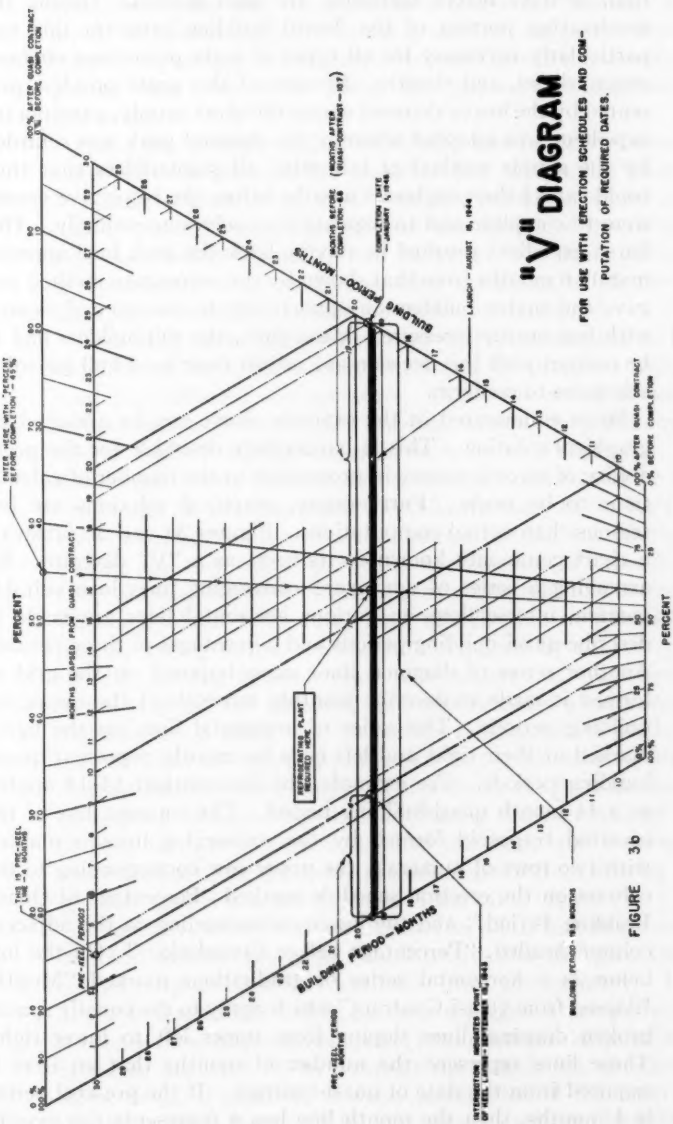

"V" DIAGRAM
FOR USE WITH ERECTION SCHEDULES AND COMPUTATION OF REQUIRED DATES.
PERCENT
MONTHS ELAPSED FROM GUARD — CONTRACT
BUILDING PERIOD — MONTHS
PRE-KEEL PERIOD
BUILDING PERIOD
LAUNCH — AUGUST 15, 1944
COMPLETION DATE — JANUARY 1, 1945
INTERMEDIATE PLANT REQUIRED HERE
ENTER HERE WITH "PERCENT BEFORE COMPLETION" = 95%
THIS IS PRE-KEEL LINE = 4 MONTHS
100% AFTER GUARD CONTRACT
0% BEFORE COMPLETION
0% AFTER GUARD CONTRACT
100% BEFORE COMPLETION

FIGURE 3b

tion, or days before launching are used instead. During the accelerating portion of the Naval building program this was particularly necessary for all types of main propulsion engines; steam, diesel, and electric. Because of this acute problem presented by the heavy demand versus the short supply, a temporary expedient was adopted whereby the demand peak was retarded by the simple method of informing all shipbuilders that they could expect their engines 3 months before the respective vessels were to complete and to regulate themselves accordingly. This harsh expedient resulted in moving back the peak load approximately 6 months from that shown by the percentage method and gave the engine builders an opportunity to tool-up and expand with less outside pressure. Meanwhile, the shipbuilders had to be content with this decision and adjust their local hull assembly schedules to conform.

Steps enumerated in the example above can be attacked by graphical solution. This is particularly desirable for the probability of error increases in proportion to the number of calculations to be made. Furthermore, graphical solutions are less tedious than actual computations. Figures 3a and 3b illustrate a chart commonly known to its users as a "V" diagram. By arranging a series of downward converging lines in a suitable pattern, intersections on various horizontal lines are made to describe quasi-building periods and percentages of those periods. Another series of diagonal lines super-imposed on the grid so formed is made to describe monthly intervals of the respective building periods. The series of horizontal lines on the figure marked at their right and left ends by months represent quasi-building periods. For example, the line marked 14–14 applies to a 14 month quasi-building period. The topmost line of the inverted trapezoid formed by the converging lines is marked with two rows of numerals, the upper row corresponding to the column on the erection schedule marked "Percentage of Quasi-Building Period," and the lower corresponding to the adjacent column headed, "Percentage before Completion." On the line below is a horizontal series of graduations marked "Months Elapsed from Quasi-Contract" which apply to the equally spaced broken diagonal lines sloping from upper left to lower right. These lines represent the number of months that an item is required from the date of quasi-contract. If the pre-keel period is 4 months, then the month line less 4 represents the months

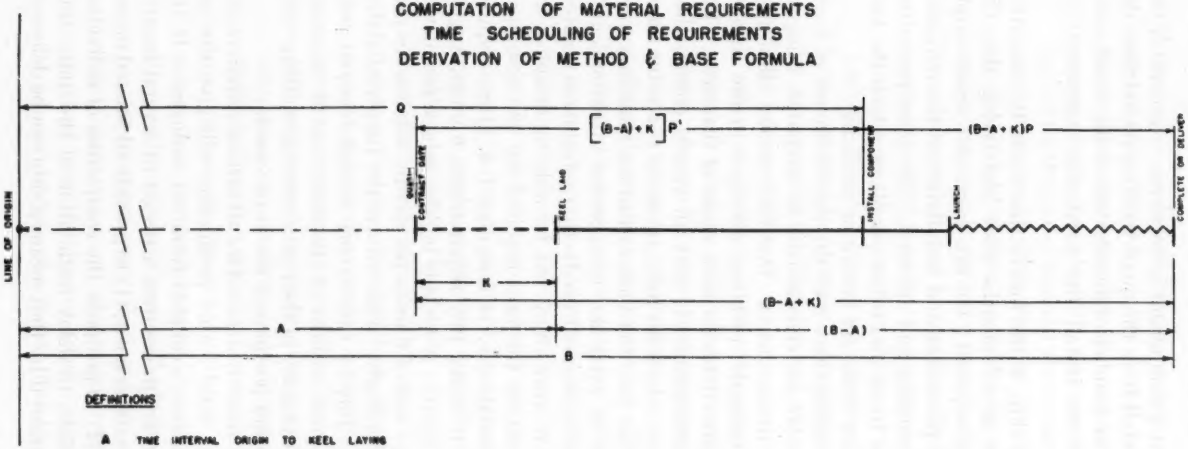
after keel laying that a component is required. Similarly if the month line is subtracted from the quasi-building period then the interval represents the number of months before the vessel completion that remain to install the particular component in question.

Using our typical ship, if the month line reads 10.7 months, then 10.7 − 4 gives us 6.7 months after keel laying that the refrigeration plant is required. On this particular vessel ample time is available for placement and installation of the refrigeration plant before launching and delivery. The same procedure is repeated for other items and other vessels on which the keel laying and completion dates are known or predicted.

Following these calculations and the establishment of basic required dates, tabular statements can be prepared using at least four different arrangements. One table shows all similar components as for example: turbines, arranged in the chronological order of required dates for each vessel of that type. The advantage of this arrangement is that all vessels receive their turbines in the order in which the hulls are ready to receive them. By separating this list into the various turbine manufacturers, leaving the turbines or particular components in chronological order a manufacturer's required production schedule is created. The manufacturer, in turn, may not be able to establish his manufacturing output at the rate required by the computed schedule as that schedule may be irregular; i. e., June requirements may call for 10 units, July requirements 6 units, August requirements 8 units, etc. A turbine builder, whose production may be at an average rate of 8 units per month, submits his production schedule accordingly. This reduces the June availability by two but increases July by two leaving the deficiency at zero. Shipbuilders are in turn notified of the delay and if necessary make allowance for the delay in their ship erection, shifting work to other vessels or other portions of the same vessel.

A summary of the monthly demand for all turbine requirements matched against the total actual production will give the net overall surplus or deficit available from the industry. If the deficit is serious and continues over a period of several months there are three alternatives open; (1) to provide additional manufacturing facilities, (2) to postpone the completion of individual ships, or (3) if possible, to delay installation of the units until the last possible moment if by that means a ship can be delivered

COMPUTATION OF MATERIAL REQUIREMENTS
TIME SCHEDULING OF REQUIREMENTS
DERIVATION OF METHOD & BASE FORMULA



DEFINITIONS

A TIME INTERVAL ORIGIN TO KEEL LAYING

B TIME INTERVAL ORIGIN TO COMPLETION

K PRE-KEEL LAYING PERIOD TO CONVERT TO ASSUMED BUILDING PERIOD.

P PERCENTAGE BUILDING PERIOD WHEN PARTICULAR COMPONENT IS REQUIRED FOR INSTALLATION IN VESSEL.

O ORIGIN OF TIME SCALE TAKEN ARBITRARILY BUT HELD CONSTANT FOR ALL VESSELS ONCE LOCATION HAS BEEN DECIDED UPON.

Q TIME INTERVAL ORIGIN TO TIME PARTICULAR COMPONENT IS REQUIRED FOR INSTALLATION IN VESSEL.

SOLUTION FOR Q $Q = \left[B-(B-A+K)P\right]$

FIGURE 4

on schedule. There is also the drawback that other components may be delayed. When this occurs, the combination may be such that there is no alternative but to postpone delivery of the vessel affected, after making due allowance for installation time of the most delayed unit and subsequent tests and trials.

The large number of components entering into ship construction and the large number of Navy ships being built require a considerable computation effort. Dates of keel laying and completion are constantly changing and time is a factor in having usable data available. For these reasons studies were made on the advisability of mechanical computing methods. This problem solved itself by the simple expedient of changing preliminary data slightly and assigning code numbers to the keel and completion dates. Punched card accounting or computing machines were then introduced to make the computing operations automatic. Figure 4 illustrates the basis of machine computation. An arbitrary base date or "line of origin" was taken as the source of all dates. Once established it remained unchanged. The degree of accuracy required in the computed dates required some decision. An error of one week plus or minus in receipt of an item will not delay a building vessel. This permits the use of 5 day intervals between successive dates instead of single day units. Thus, if a main feed pump is required on June 17, 1944, no damage or delay will be caused by its receipt on June 15, or June 20.

After decision as to date intervals, a simple date table was prepared using the base date as zero and every succeeding five day interval thereafter as an additional unit increment. In this manner, by designating December 25, 1938, as "0", January 5, 1939, as "2," January 10, as "3," etc., and disregarding the small error caused by the 31 day and 28 or 29 day months, a date code was prepared. Figure 5 is a "Date Code Number" table prepared in this manner. All that now remained was to convert the reported keel date and completion date into their respective numbers, punch these into suitable cards and feed them into the machines. The machines when properly set up and connected apply the appropriate pre-keel intervals, compute the keel-completion interval, and from these, the quasi-building period. By a judicious selection of master cards and pre-computed percentages the machines in turn computed the actual coded date a component was required. A further step translated

DATE CODE NUMBERS
(TO BE USED WITH 6X FACTOR)

DATE	CODE NO.	DATE	CODE NO.	DATE	CODE NO.	DATE	CODE NO.	DATE	CODE NO.	DATE	CODE NO.
12-31-38	001	12-31-39	073	12-31-40	145	12-31-41	217	12-31-42	289	12-31-43	361
1-8-39	002	1-5-40	074	1-5-41	146	1-5-42	218	1-5-43	290	1-5-44	362
1-10-39	003	1-10-40	075	1-10-41	147	1-10-42	219	1-10-43	291	1-10-44	363
1-15-39	004	1-15-40	076	1-15-41	148	1-15-42	220	1-15-43	292	1-15-44	364
1-20-39	005	1-20-40	077	1-20-41	149	1-20-42	221	1-20-43	293	1-20-44	365
1-25-39	006	1-25-40	078	1-25-41	150	1-25-42	222	1-25-43	294	1-25-44	366
1-31-39	007	1-31-40	079	1-31-41	151	1-31-42	223	1-31-43	295	1-31-44	367
2-5-39	008	2-5-40	080	2-5-41	152	2-5-42	224	2-5-43	296	2-5-44	368
2-10-39	009	2-10-40	081	2-10-41	153	2-10-42	225	2-10-43	297	2-10-44	369
2-15-39	010	2-15-40	082	2-15-41	154	2-15-42	226	2-15-43	298	2-15-44	370
2-20-39	011	2-20-40	083	2-20-41	155	2-20-42	227	2-20-43	299	2-20-44	371
2-25-39	012	2-25-40	084	2-25-41	156	2-25-42	228	2-25-43	300	2-25-44	372
2-28-39	013	2-29-40	085	2-28-41	157	2-28-42	229	2-28-43	301	2-29-44	373
3-5-39	014	3-5-40	086	3-5-41	158	3-5-42	230	3-5-43	302	3-5-44	374
3-10-39	015	3-10-40	087	3-10-41	159	3-10-42	231	3-10-43	303	3-10-44	375
3-15-39	016	3-15-40	088	3-15-41	160	3-15-42	232	3-15-43	304	3-15-44	376
3-20-39	017	3-20-40	089	3-20-41	161	3-20-42	233	3-20-43	305	3-20-44	377
3-25-39	018	3-25-40	090	3-25-41	162	3-25-42	234	3-25-43	306	3-25-44	378
3-31-39	019	3-31-40	091	3-31-41	163	3-31-42	235	3-31-43	307	3-31-44	379
4-5-39	020	4-5-40	092	4-5-41	164	4-5-42	236	4-5-43	308	4-5-44	380
4-10-39	021	4-10-40	093	4-10-41	165	4-10-42	237	4-10-43	309	4-10-44	381
4-15-39	022	4-15-40	094	4-15-41	166	4-15-42	238	4-15-43	310	4-15-44	382
4-20-39	023	4-20-40	095	4-20-41	167	4-20-42	239	4-20-43	311	4-20-44	383
4-25-39	024	4-25-40	096	4-25-41	168	4-25-42	240	4-25-43	312	4-29-44	384
4-30-39	025	4-30-40	097	4-30-41	169	4-30-42	241	4-30-43	313	4-30-44	385
5-5-39	026	5-5-40	098	5-5-41	170	5-5-42	242	5-5-43	314	5-5-44	386
5-10-39	027	5-10-40	099	5-10-41	171	5-10-42	243	5-10-43	315	5-10-44	387
5-15-39	028	5-15-40	100	5-15-41	172	5-15-42	244	5-15-43	316	5-15-44	388
5-20-39	029	5-20-40	101	5-20-41	173	5-20-42	245	5-20-43	317	5-20-44	389
5-25-39	030	5-25-40	102	5-25-41	174	5-25-42	246	5-25-43	318	5-25-44	390
5-31-39	031	5-31-40	103	5-31-41	175	5-31-42	247	5-31-43	319	5-31-44	391
6-5-39	032	6-5-40	104	6-5-41	176	6-5-42	248	6-5-43	320	6-5-44	392
6-10-39	033	6-10-40	105	6-10-41	177	6-10-42	249	6-10-43	321	6-10-44	393
6-15-39	034	6-15-40	106	6-15-41	178	6-15-42	250	6-15-43	322	6-15-44	394
6-20-39	035	6-20-40	107	6-20-41	179	6-20-42	251	6-20-43	323	6-20-44	395
6-29-39	036	6-25-40	108	6-25-41	180	6-25-42	252	6-25-43	324	6-25-44	396
6-30-39	037	6-30-40	109	6-30-41	181	6-30-42	253	6-30-43	325	6-30-44	397
7-8-39	038	7-5-40	110	7-5-41	182	7-5-42	254	7-5-43	326	7-5-44	398
7-10-39	039	7-10-40	111	7-10-41	183	7-10-42	255	7-10-43	327	7-10-44	399
7-15-39	040	7-15-40	112	7-15-41	184	7-15-42	256	7-15-43	328	7-18-44	400
7-20-39	041	7-20-40	113	7-20-41	185	7-20-42	257	7-20-43	329	7-20-44	401
7-25-39	042	7-25-40	114	7-25-41	186	7-25-42	258	7-25-43	330	7-25-44	402
7-31-39	043	7-31-40	115	7-31-41	187	7-31-42	259	7-31-43	331	7-31-44	403
8-5-39	044	8-5-40	116	8-5-41	188	8-5-42	260	8-5-43	332	8-5-44	404
8-10-39	045	8-10-40	117	8-10-41	189	8-10-42	261	8-10-43	333	8-10-44	405
8-15-39	046	8-15-40	118	8-15-41	190	8-15-42	262	8-15-43	334	8-15-44	406
8-20-39	047	8-20-40	119	8-20-41	191	8-20-42	263	8-20-43	335	8-20-44	407
8-25-39	048	8-25-40	120	8-25-41	192	8-25-42	264	8-25-43	336	8-25-44	408
8-31-39	049	8-31-40	121	8-31-41	193	8-31-42	265	8-31-43	337	8-31-44	409
9-5-39	050	9-5-40	122	9-5-41	194	9-5-42	266	9-5-43	338	9-5-44	410
9-10-39	051	9-10-40	123	9-10-41	195	9-10-42	267	9-10-43	339	9-10-44	411
9-15-39	052	9-15-40	124	9-15-41	196	9-15-42	268	9-15-43	340	9-15-44	412
9-20-39	053	9-20-40	125	9-20-41	197	9-20-42	269	9-20-43	341	9-20-44	413
9-25-39	054	9-25-40	126	9-25-41	198	9-25-42	270	9-25-43	342	9-29-44	414
9-30-39	055	9-30-40	127	9-30-41	199	9-30-42	271	9-30-43	343	9-30-44	415
10-9-39	056	10-5-40	128	10-5-41	200	10-5-42	272	10-5-43	344	10-5-44	416
10-10-39	057	10-10-40	129	10-10-41	201	10-10-42	273	10-10-43	345	10-10-44	417
10-15-39	058	10-15-40	130	10-15-41	202	10-15-42	274	10-15-43	346	10-18-44	418
10-20-39	059	10-20-40	131	10-20-41	203	10-20-42	275	10-20-43	347	10-20-44	419
10-25-39	060	10-25-40	132	10-25-41	204	10-25-42	276	10-25-43	348	10-25-44	420
10-31-39	061	10-31-40	133	10-31-41	205	10-31-42	277	10-31-43	349	10-31-44	421
11-5-39	062	11-5-40	134	11-5-41	206	11-5-42	278	11-5-43	350	11-5-44	422
11-10-39	063	11-10-40	135	11-10-41	207	11-10-42	279	11-10-43	351	11-10-44	423
11-15-39	064	11-15-40	136	11-15-41	208	11-15-42	280	11-15-43	352	11-15-44	424
11-20-39	065	11-20-40	137	11-20-41	209	11-20-42	281	11-20-43	353	11-20-44	425
11-25-39	066	11-25-40	138	11-25-41	210	11-25-42	282	11-25-43	354	11-23-44	426
11-30-39	067	11-30-40	139	11-30-41	211	11-30-42	283	11-30-43	355	11-30-44	427
12-5-39	068	12-5-40	140	12-5-41	212	12-5-42	284	12-5-43	356	12-5-44	428
12-10-39	069	12-10-40	141	12-10-41	213	12-10-42	285	12-10-43	357	12-10-44	429
12-15-39	070	12-15-40	142	12-15-41	214	12-15-42	286	12-15-43	358	12-15-44	430
12-20-39	071	12-20-40	143	12-20-41	215	12-20-42	287	12-20-43	359	12-20-44	431
12-29-39	072	12-25-40	144	12-25-41	216	12-25-42	288	12-25-43	360	12-25-44	432

FIGURE 5

the coded date into its calendar date for the respective card and vessel.

The next steps sorted the individual cards into types of vessels by percentages or lead times and printed them either chronologically or consecutively by vessel designation numbers. Figure 6 is a reproduction of the latest tabulation form. Note that it bears two distinct listings, the first by consecutive vessel or Navy hull numbers, the second in chronological order of required dates. The two arrangements are supplementary and facilitate finding of data. A series of the forms so prepared and classified into sections by percentages, lead times, (in the case of days before completion) and further separated in vessel classes, is bound into a volume and distributed as a COMPONENT PERCENTAGE SCHEDULE. The SCHEDULE is generally known to its users as the "CPS." With this tool in their hands, and also the MATERIAL ERECTION SCHEDULES (MES), shipbuilders, manufacturers, Naval inspection activities, and others can quickly determine when certain materials or components are required for shipment. Note that the term "required for shipment" is used instead of "delivery date." The COMPONENT PERCENTAGE SCHEDULE as published has a correction factor of 15 days which is subtracted from the delivery date as computed. The use of a correction factor was found advisable as the major use of the schedule was applied to shipments from parts and component manufacturers in the chain of supply, 15 days representing an average transit time from producer to user.

Originally, the plan contemplated three series of forms. The first two consisted of tabulated required dates for each component that the Bureau of Ships intended to schedule. One form listed vessels consecutively by hull numbers, and the other listed required dates in chronological order similar to the present COMPONENT PERCENTAGE SCHEDULE. A third form listed all principal components for vessels under individual shipbuilders. The forms were further arranged so that after removal from the tabulating printers, they could be sorted out and sent to the respective component manufacturers as shipping instructions and permit the manufacturers in turn to submit their promised dates against the computed required dates. However, the original plan was never instituted primarily due to a decentralization of scheduling effort in the Bureau itself, the desire to place a simpler

NavSHIPS 17351
RESTRICTED

ARRANGED BY HULL NUMBER

1 JULY 1944

PRECEDENCE NO	VESSEL TYPE	VESSEL NO	SHIPBUILDER	REQUIRED SHIPMENT DATE	%	END OF CONSTRUCTION DATE
1670	ABC	10	SHPBLDR INC	2 14 44	45	12 5 44
1690	ABC	11	SHPBLDR INC	4 4 44	45	2 5 45
1655	ABC	12	TANKER CORP	2 6 44	45	11 1 44
1685	ABC	13	TANKER CORP	4 6 44	45	1 1 45
1693	ABC	14	TANKER CORP	6 6 44	45	3 1 45
1620	XYZ	3	JONS & JONS	2 20 44	45	7 5 44
1634	XYZ	4	JONS & JONS	4 5 44	45	8 20 44
1648	XYZ	5	JONS & JONS	5 21 44	45	10 5 44
1644	XYZ	7	SMITH CO	3 28 44	45	9 10 44
1677	XYZ	8	SMITH CO	7 7 44	45	12 25 44

ARRANGED BY HULL NUMBER

NavSHIPS 17351

ARRANGED BY REQUIRED ORDER

RESTRICTED
1 JULY 1944

PRECEDENCE NO	VESSEL TYPE	VESSEL NO	SHIPBUILDER	REQUIRED SHIPMENT DATE	%
1655	ABC	12	TANKER CORP	2 6 44	45
1670	ABC	10	SHPBLDR INC	2 14 44	45
1620	XYZ	3	JONS & JONS	2 20 44	45
1644	XYZ	7	SMITH CO	3 28 44	45
1690	ABC	11	SHPBLDR INC	4 4 44	45
1634	XYZ	4	JONS & JONS	4 5 44	45
1685	ABC	13	TANKER CORP	4 6 44	45
1648	XYZ	5	JONS & JONS	5 21 44	45
1693	ABC	14	TANKER CORP	6 6 44	45
1677	XYZ	8	SMITH CO	7 7 44	45

ARRANGED BY REQUIRED ORDER

FIGURE 6

working tool in the hands of inspectors and industry, and a decentralization to the field inspection services of the Bureau.

With the shipping dates now available and on a standard basis, shipbuilders and ordering agencies are charged with the placing of correct delivery dates on their purchase orders. One pronounced result has been the stoppage of the tendency to inflate a manufacturer's production schedule. This tendency gave the manufacturer a false impression of the actual demand rate and caused him to either add additional production facilities or to appear as a "bottleneck" in the production picture. By eliminating this condition, true constrictions in the production program have been shown and manufacturing facilities were furnished where most needed.

Supplementing the "MATERIAL ERECTION SCHEDULE" is a MATERIAL ERECTION INDEX. The erection schedule lists, as stated before, the principal items entering into the construction of a vessel. There are in addition many more items entering into a vessel, principally as part and subcomponents of the principal ones, for a ship is probably the most complicated single material creation of man. In addition there is confusion of terminology caused by local shipyard usage and also by the language peculiar to the sea differing from that used on land. The difference is most marked where manufacturers who have previously built only for shore use and have converted to marine use are required to learn a new tongue. There is the difference between "smokepipes" and "stacks," "bits and bollards," "ladders" and "stairs," "winches" and "hoists" to cite just a few cases. For these reasons a definitive list of terms was created to familiarize the user of the MATERIAL ERECTION SCHEDULE with the general usage of terms and also to spot various otherwise unlisted details into their proper location on the schedule.

As has been stated, one of the original intents was to have all manufacturers submit their promised delivery dates. The original conception was rejected because of the mass of detail involved and conflicts with the allocation of bulk purchased products which were not directly assignable to specific vessels. Part of the answer was supplied by the War Production Board in a special regulation which required that certain production forms be submitted, showing anticipated production rates on selected lists of critical products. This form had originally been devised

by the Bureau of Ships to obtain production output data from manufacturers, but was quickly adopted in a slightly modified form by the War Production Board for use by all government agencies concerned with manufacturers' production data. It is received monthly by the Bureau on products the Bureau schedules directly and when completely filled out shows individual manufacturers' anticipated production, the Bureau's requirements, and actual shipments for a ceries of monthly intervals. The Bureau's requirements are, of course, worked up from the Component Percentage and Material Erection Schedules plus such additional quantities that may be required for repair purposes.

The production schedules so obtained when summarized for a single class of product and for the entire industry reveal products which are critical in procurement for the summaries show supply matched against demand. They also show surplus capacity available for diversion to other product manufacture or for additional repair part manufacture. After being reviewed, the production schedules are adjusted to show the Bureau's actual needs which are a combination of new ship construction requirements and replacements for similar units lost in service operations. From here the schedules are returned to the manufacturer as his "directive" to produce at the rate assigned.

Certain minor modifications have been made to the scheduling procedure where fast moving shipbuilding programs were injected into the picture such as the escort vessel program and the landing craft program. To assist the shipbuilders in having material available in sufficient time to meet any impact of possible delay on specific items, modified erection schedules were substituted for detailed percentage schedules. These modified schedules are known as "Package Schedules." Instead of listing separate percentages opposite individual items, blocks of items have been consolidated into a single grouping or "package" under one percentage figure. For example: Material Erection Schedule items 1 to 15 inclusive may be grouped into Package 1, using as an availability indicator, 85 per cent before completion. By this arrangement a Material Erection Schedule, consisting of sixty items, may be reduced to four packages. The advantage of this procedure lies primarily in the reduction of the number of dates to be listed for follow-up purposes, and the earlier availability of certain components to a shipbuilder.

The "rush" programs also received extra attention in that special expediting agencies were established in key cities for the purpose of directly progressing all production phases from plans to material delivery, including the various products and items that went into the ships. Best known of such offices was the Material Coordinating Agency, Inc., in New York, familiarly known as MCA, operating directly as a branch of the Bureau of Ships. These agencies established their own expediting methods, forms, and procedures, using the Bureau material erection schedules and component percentage schedules as their basic guides for materials being delivered to the competing shipyards.

Much of the success of the scheduling procedure has also been due to an intensive course of instruction given to the Navy's field inspection personnel and actual field instruction by representatives of the Bureau of Ships. Cooperation of shipbuilders and manufacturers has in most cases been complete and voluntary. For recalcitrant individuals, enforcement has been possible by the Bureau's requirement that any and all orders placed for shipbuilding materials carry as their delivery dates those listed in the latest Component Percentage Schedule. The right has also been assigned to the Bureau's inspection forces to make such changes where the purchaser has not complied. Further force has been given to the Bureau of Ships ability and right to change any delivery dates by delegations of authority from the War Production Board. To this legal authority plus the wholehearted cooperation of industry in applying the Scheduling Plan, goes part of the credit which has caused our war vessels to be delivered at a rate beyond that originally thought possible in 1940 when the first of the big shipbuilding contracts were awarded.

COMMON SENSE PRACTICES IN THE USE OF WELDING IN STEEL SHIP CONSTRUCTION.

By A. G. Bissell.*

To use welding successfully in steel ship construction, it is necessary that the many factors entering into this form of construction be recognized, understood and properly administered. In too many instances, a shipyard whose personnel were brought up on riveted practices has embarked into the use of welding without serious consideration of the factors involved in such work with the result that unsatisfactory work has been obtained and the use of welding condemned.

It is necessary that the advisory, supervisory and operating force be thoroughly familiar with the use of welding, its advantages and limitations. This group should be coordinated so that they function together as one unit and not as independent departments. If this is done, it will insure that the proper designs, procedures and sequences are used and a product of maximum strength and durability will be produced. The success or failure of the use of welding in shipbuilding hinges on the smooth coordination of the operating, supervisory and advisory forces.

The advisory force should consist of the welding engineer and his assistants. They should advise as to the welding design, the welding procedure and welding sequence. The latter will, in a great measure, determine the erection sequences. The welding engineer's group in conjunction with the supervisory heads should prepare definite procedures and sequences for the erection and fabrication of the work to be done. No procedure or sequence should be issued to the operating force until it has been agreed to and approved by the welding engineer and the supervisory heads. Once issued, these instructions should not be changed or modified without the complete accord of this group. In the application of these instructions, the welding engineer and his assistants should carefully check the results obtained and any diversion from the expected results should be noted, the probable cause determined

*Bureau of Ships, Navy Department.

and corrective measures instituted after being approved by all concerned. Without such close cooperation, the success of welded ship construction cannot be fully realized.

A prime requisite in welded ship construction is that the design be correct. All connections should be made so as to avoid sudden changes in section and no sharp cornered openings or inserts should be employed. The foregoing is probably the most pertinent advice that can be given to the designer. Excess of welding in a specific location should be avoided and all joints should be so designed and placed to afford complete and easy access in order that complete fusion can be obtained throughout the weld. Where complete access is possible from one side only, the design should indicate the use of a backing strip and not rely on the operating force to decide on this expedient. In consultation with the welding engineer, the most economical joint design should be decided upon and included in the plans. The size of fillet welds necessary to develop the required efficiency should be determined and indicated, as electrode economy and production speeds are dependent on this factor. The amount of weld metal in fillet welds increases about as the square of the throat dimensions, hence an oversized fillet weld can be a definite disadvantage. The design should permit fabrication of sub-assemblies which should be as large as can be conveniently handled, thus permitting the use of positioning equipment and the so-called "hot rods" and materially increasing the rate of production over that obtained in using stationary work and the reverse polarity all-position electrodes.

Each step of the erection and welding sequence should be carefully planned in order that too much steel will not be erected ahead of the production welding. This is a common error in many yards with the result that the material being welded is not free to move as it shrinks under the welding heat with the result that the bow or stern usually rises off the keel blocks or fractures occur in the restrained plating. This may be avoided by tacking not more than one assembly section in advance of the production welding. Also by using tacks that will fracture and not heavy intermittent welds. When a tack weld cracks, adjustment should be made to compensate for the shrinkage and a new tack installed. The use of a welding sequence which requires working out from a neutral center to free any unrestrained ends usually gives the minimum of distortion. In this outward pro-

gression, a back step sequence is desirable. However, the block sequence tends also to give a uniform result. In either case, the deposit weld metal upon cooling is just about at its yield point but uniformly loaded when either back step or block sequence is used. However, it is possible in using a straight forward progression to build up stresses that will exceed the ultimate tensile strength of the steel which upon cooling will result in fractures.

The effective ultimate strength of the steel used in shipbuilding may be decidedly lowered by the presence of discontinuities, hence the earlier admonition in regard to sudden changes of sections and avoidance of sharp corners. These discontinuities reduce the resistance of the steel to impact especially at low temperatures. The notched impact value of a normal medium carbon steel at 20 degrees F. may be but about one-fifth to one-seventh that of the same steel at 70 degrees F. Hence, it is most desirable to eliminate all notches and square corners where a fracture might start. The upper edge of the sheer strake, being a highly stressed edge when the vessel is hogged, should be kept as free as possible from attachments, especially at the three-fifths midship length. Railing stanchions, fair leads, accommodation ladder connections, etc., should be attached to the stringer strake inboard from the edge of the sheer strake. Such attachment should be made by smooth welds, free from undercutting. Hatch openings and hatch coamings should have well rounded corners and the top edges of the coaming should be well rounded and free from gas cut nicks. It is the writer's opinion that the attachment of a solid half-round to the edge of the sheer strake or hatch coaming is more detrimental and not as desirous as the unreinforced edges well rounded and smooth. The same advice is given in regard to the corners of deck houses; they should be well rounded. Where a superstructure fairs into the sheer strake, a well curved fashioned plate of equal thickness should be used which connects to the edge of the sheer strake with a smooth and unbroken connection. Angular holes such as scuppers should be avoided in the upper edge of the sheer strake. If drains are to be provided, they should be constructed of pipe piercing the stringer and sheer strakes through round holes clear from the plate edges. The inserted pipe should be completely and smoothly welded into place. Difficulty has been experienced where scuppers have been cut through the sheer strake by gas cutting, leaving a ragged and

unfinished cut. Any cut made in a strength member of a welded vessel or any other vessel, in fact, should be well smoothed and rounded to eliminate stress raisers that might be the starting point of a crack. A crack once started in a welded vessel will continue as long as it can progress into a stressed area. A weld transverse to a progressing crack offers no resistance to the crack since the weld is the same general structure as the plate. To control cracking in vessels, it is necessary to eliminate the starting point by eliminating stress raisers in highly stressed members. And this in turn is controlled by proper design to eliminate the stress raisers, proper erection and welding sequence to reduce residual stresses and the use of electrodes that give solid homogeneous welds of maximum ductility and a yield and tensile strength of slightly over that of the plate material. It has been found that a welded joint made with a high ductility weld material has the lowest residual stress.

In the fabrication of welded ships, we must consider the shop organization. In most yards, the master shipfitter is the ruling head and in many cases, he is an old-timer that has come up through the school of riveted ships. In his organization, of course, he favors the shipfitter supervisors, who in turn dominate the welding supervisory force. In such yards, the advance of welded construction has been very slow and satisfactory results limited as there has been a lack of understanding of the necessity of good fitting and proper joint preparation because, in the early days, welding was generally used to correct errors and misfits, so it is not unusual for the welder, when requesting a better fit for a joint to be welded, to be told that he is a poor welder if he cannot weld up such a joint. It is the writer's opinion that many of the so called weld slugging cases that have been given much publicity in the last couple of years have been more the result of poor fitting and lack of proper supervision than the operator's desire to increase his production and pay. Obviously, had the welding operator been given a properly fit-up joint which he could have welded and made his piecework rate on, there would have been no reason for him to lay in electrodes to fill up an oversized gap.

The ideal organization is one in which the supervisory head and both his erection foreman and welding foreman are well trained in the requirements of welded construction and are working to the single goal of a satisfactory and well constructed

ship. Such an organization working under an approved erection and welding sequence using the correct joint design and welding materials will have a minimum amount of weld metal to deposit and less distortion or locked up stresses in the finished ship.

In the construction of welded ships, satisfactory welding apparatus should be used whether it be for arc welding or gas welding when it is used. Since arc welding is the most extensively used, I shall comment more on that phase. The welding equipment whether it be AC or DC, multiple operator or single operator, should be in a good state of repair and have amply fine adjustment to give the operator the control necessary for good welding. Each of the foregoing equipment has a useful place in the industry and should be applied under the direction and advice of the welding engineer. The welding leads should be kept in good condition and systematically strung over the job. It is very desirable to have the leads broken down into lengths of 50 or 100 feet and fitted with suitable connectors which permit the leads to be disconnected and reconnected in unsnarling tangles, also this permits the stringing of new leads to other locations and interchanging with the lead that is being used on a job that has just been completed, thus utilizing a welder's time on welding rather than pulling cable. The cable should be of ample size to reduce loss of power and prevent overheating. 0 and 00 cables usually are ample for most applications. The electrode holders that have been used in the past have not been of the best design. An electrode holder should be of such construction that it tightly grips the electrode ends yet permits the electrodes to be used to minimum stub lengths. It should be so insulated that in melting the electrodes down to this minimum, it will not short circuit against the work. It should also be so insulated that if the operator accidentally falls or otherwise brings the holder in contact with the body, there is no opportunity for his body to become grounded to the welding circuit. This is especially necessary when welding operators are working in confined spaces in hot weather.

Ventilation for the removal of welding fumes and generally supplying fresh air to confined spaces is imperative and should not be neglected. Even in open spaces, provision should be made for the removal of fumes when welding is done on galvanized surfaces or where other harmful fumes may result from the composition of the materials involved in the welding process.

Wherever possible, positioning equipment should be used to place the work in the flat position to permit the use of the maximum size of the so-called "hot rods" for either butt or fillet welding, thus increasing the speed of production and reducing distortion and residual stresses to a minimum.

In erection on the building ways, care should be taken to complete the welding throughout the entire cross section of the vessel as much as possible as the fabrication progresses toward free ends, thus obtaining smooth and uniform keel and shell lines and eliminating the necessity of doing any great amount of welding above the neutral center after launching. There have been cases where a vessel was launched as soon as the hull would float and considerable welding done on the decks and superstructure after launching with the result that the shaft struts were well out of line due to the shrinkage of the upper portion causing the vessel to sag. In all cases an excess of material should be provided on the free ends of all plates to compensate for the shrinkage which occurs due to the heat of welding, this and a balanced progression from the welding center outward and fore and aft are necessary to obtain a true structure with a minimum of residual stresses. Shell butts should be welded as reached and always completed when once started. Shell seams if stopped for any reason should be built up to the full size of the weld and not left with a tapered off weld ending in a crater. This practice and partially welding butts have been responsible for many construction fractures, especially in cold weather and between shifts. The so-called washboard distortion can be reduced to a great extent if a slight clearance is left between the frames and the shell plating to permit the shrinkage of the deposited weld metal to pull up the frames instead of bending the shell plating. This is also desirable in the attaching of stiffeners to bulkheads and imperative in joining the vertical to the flat keel. This spacing may be accomplished by using soft iron wire spacers for the lighter members or by rough machining the edge of the abutting member of the heavier sections. The excessive use of strongbacks and erection clips should be avoided as the attachment of these appendages tends to increase the accumulation of residual stresses and results in damage to the surface of the plating forming potential starting points of cracks.

After the vessel is launched, we must protect the underwater body from electrolytic corrosion if we are on salt water by

the proper grounding of the welding leads in such a manner as to avoid any opportunity of the welding current passing from hull to hull or from hull to shore. Even where the work is being done in fresh water, the same care should be exercised as most fresh water contains sufficient industrial waste to make it a good conductor. In some construction where the outfitting time is not too long, it has been found an advantage to eliminate the painting of the hull prior to launching, thus presenting an unlimited area for stray currents to enter and leave the hull whereas on a painted hull that has been marred in launching, scratches permit a concentrated attack resulting in deep line corrosion. The safest procedure, however, is to ground the welding sets to the vessels on which they are being used and not to weld on any other vessel unless the ground lead has been disconnected from the first vessel and changed to the one on which the work is to be done. Multiple operator sets should not be used on waterborne vessels unless all the leads from such sets are used upon one vessel to which the set is grounded. Multiple operator sets generally should be used in the shops or on the building ways and single operator sets used on waterborne vessels, except when all of the outlets of the multiple operator's sets can be confined to a single vessel.

The foregoing has been a general review of the factors that must be considered if satisfactory results are to be obtained in welded ship construction. There are many details which have not been mentioned which must be considered by the welding engineer as the work progresses, such as preheating, stress relieving, preforming to avoid distortion, the selection of electrodes for special purposes, etc. Of all the factors that enter into the work, the erection and welding sequences and correct joint preparation probably are the most pertinent features. If these are carefully considered and enforced, the resulting structure should be free from excessive locked up stresses concentrated at points to start failures which might cause loss of the vessel. After all, it is just the application of plain common sense to design and construction of the ship which is necessary to produce a good job.

The comments of the foregoing article are the opinions of the author and do not necessarily represent the opinions of the Bureau of Ships or the Navy Department.

SPARKING AND FIRES IN DIESEL ENGINE EXHAUST STACKS.

By Lt. Commander Ralph Baggaley, U. S. N. R.

A reporter covering Vesuvius in action does not present a more vivid account of exciting pyrotechnics than is usually given by shipboard personnel in describing their Diesel stack fires and related occurrences. The similarity between the two happens to be rather marked, as the stack, like Vesuvius, can reach a glowing red heat, explode, or eject either large or small incandescent material well up into the air. That the sight is inspiring is attested to by engineers not given to exaggeration reporting, "Great Balls of Fire came out of the Stack."

While lacking neither in color nor force, many reports show lack of understanding of causes as well as of feasible corrective measures. The stack phenomena are particularly serious on tankers because of the added fire hazard, and on all vessels proceeding in hostile waters at night, because of the high visibility entailed.

The data herein is presented for the benefit of those charged with motorship maintenance and/or operation, to provide a clearer picture of what preceded the stack eruptions and of how to avoid them.

A typical marine Diesel exhaust system is shown in Figures 1 and 2. The very considerable length of this system (60 feet in an average 6000 horsepower motorship) is subject to coating with whatever oil or sooty carbon may exist in the exhaust gases leaving the engine. Such deposition generally occurs during periods of light engine loading. When high speed operation is undertaken, greatly increased heat and gas velocity is impressed upon the system which ignites and ejects the previously deposited oil and soot.

The various types of phenomena and the sequence of events leading up to them are usually as follows:

(*a*) *Fires and Explosions:* Generally occur after engines have been operating at low load with excess lubricating oil in the cylinders. Under these conditions unburned lube oil is blown

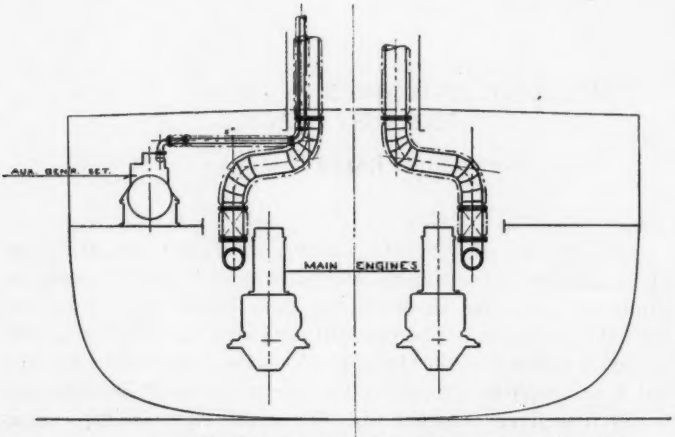


FIGURE 1 - SECTION LOOKING AFT TO MAIN ENG. EXHAUSTS.

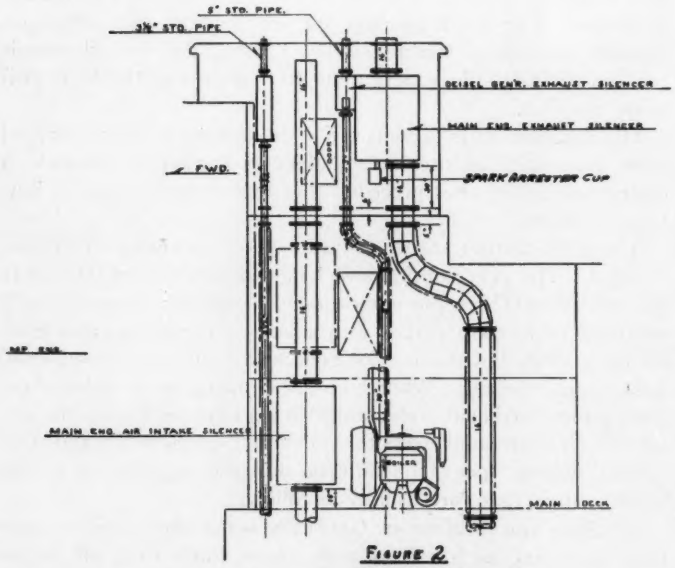


FIGURE 2.
ELEVATION
LOOKING TO STBD.

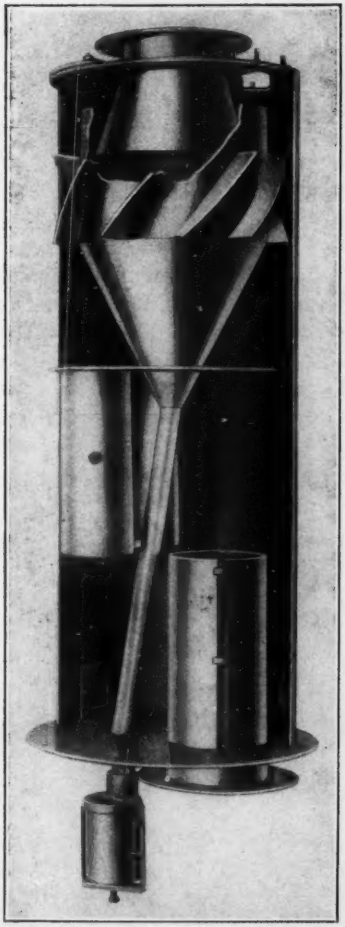


FIGURE 3.—MAXIM SPARK ARRESTER SILENCER.

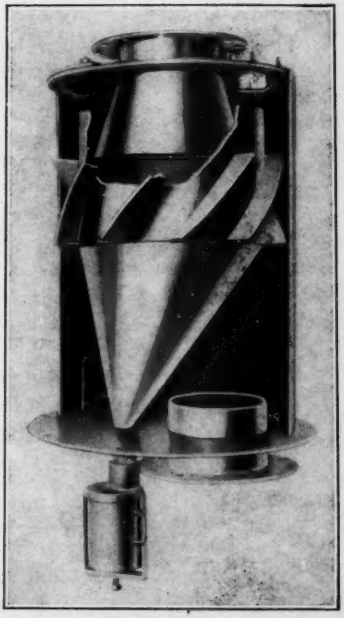


FIGURE 4.—MAXIM SPARK ARRESTER.

into exhaust passages resulting in accumulations of oil on horizontal surfaces in the piping or muffler as well as a coating of oil throughout the exhaust system. When engine loading is increased, the higher exhaust temperature ignites the oil in the system, resulting in a fire or an explosion.

(*b*) *Sparking:* Operation of engines at low speed and light load has coated interior of the exhaust system with soft carbon. As operating hours mount up, under layers of carbon are baked hard, while additional soft carbon is deposited on top. When engine loading is increased with resultant higher exhaust temperature and pressure, pieces of hard carbon are burnt free and are blown out of the stack in an incandescent state. This condition is known as "Sparking." When incandescent small particles of the soft carbon without pieces of hard carbon are being ejected, the condition is referred to as "Torching." It occurs under conditions similar to those causing sparking, but also frequently happens without increased engine loading.

From the above it is apparent the problem is primarily one of eliminating as far as possible, lube oil and carbon from the exhaust system, and secondarily one of controlling the effect of such lube oil and carbon as may remain.

Equipment required for checking the subject conditions is as follows:

(*a*) A drain line from a low point in the muffler or piping to remove oil as it starts to accumulate before fire or sparks from subsequent higher speed operation touch it off.

(*b*) At least one clean-out opening in the exhaust system. In systems of merchant vessel size this should be a 14 inch x 16 inch or larger manhole in the muffler shell and a readily removable short section of exhaust pipe midway between the engine and muffler.

(*c*) An efficient spark arrester, either incorporated in the muffler or provided as a separate unit may be necessary to prevent sparking from unavoidable small accumulations of carbon. Examples of each type arrester are shown in Figures 3 and 4 respectively. Although not included in the illustrations, an oil drain from the low point of each silencer, ending in a receptacle convenient for periodic draining should be incorporated.

The writer does not favor the use of various proprietary chemicals purporting to overcome fires and sparking. Certain of these are for mixing into the fuel oil and others for injection

into the exhaust system. Several have been tested and found to be without merit.

Following is a list of recommended operating procedures:

(*a*) Keep lube oil in the exhaust system at a minimum:

(1) By avoiding excess feed through mechanical cylinder lubricator. Do not reduce lubricator feed below amount specified by engine manufacturer.

(2) By prompt replacement of worn piston oil control rings.

(3) By checking crankcase breather connections to the air induction system, if the breathers are suspected of passing oil.

(4) By checking for leaks from lube oil seals on blowers.

(*b*) Maintain fuel injection equipment in good condition to avoid smoky exhaust with resultant deposition of carbon in the system.

(*c*) As indicated by experience with the particular installation, provide a schedule for periodically emptying the oil drains installed in the system.

(*d*) At the end of each run when practicable examine interior of exhaust system for accumulations of carbon, and remove if found. This should include the exhaust ports of the engines, the exhaust piping and the mufflers.

(*e*) As far as practicable, operate engines at loadings which will produce a clear exhaust. Keep duration of engine warm-up periods without load to the minimum required.

(*f*) If operation at unfavorable speeds is necessary and the vessel is in hostile waters, endeavor to operate for one-half hour at increased speed before nightfall in order to burn out any slight excess of carbon before darkness sets in.

In conclusion, it should be stated that the requirements for avoiding the various stack phenomena are first, proper exhaust equipment; second, properly engineered and maintained engines; and third, the ever present shipboard necessity of good housekeeping.

THE EFFECT OF LOCATION UPON TORSION METER READINGS.

By Hugh P. Webster, Commander, U. S. N.
and Frank A. Scanlan.*

Prior to the present emergency the data for the following article was compiled during a routine shop test but was not published. Because of the appearance of the excellent treatise "Accuracy of Shaft Horsepower" by Mr. Harold L. Coffin, in the February 1944 issue of the Journal of the American Society of Naval Engineers, it is believed that this supplementary information would at this time enhance the professional aspect of the subject.

Until a few years ago, the measurement of the torque factor for main propulsion shafting power transmission, for trial purposes, was accomplished by means of a meter which was entirely mechanical in construction and operation. Because this instrument was always inserted between the flanges of a specially calibrated length of shafting, no problem as to the effect of location upon the readings of the meter was encountered. The readings of this mechanical meter gave the average twist for the entire length of the special shaft.

The purely mechanical type torsion meter has been superseded for Naval use by an electrical type which is of such design that it can be fitted externally upon the shafting at any convenient location. The length between the clamping planes is less than 3 feet, giving considerable flexibility as to choice of location in the average vessel. The principles of this meter are described in the article referred to in the first paragraph.

A mental comparison of the two types mentioned above raises the question of the effect the location will have upon the readings obtained from the electrical torsion meter. The length of this instrument precludes it from giving an average value of twist for other than the comparatively short length of the shaft between the clamping planes. The location selected for this external type torsion meter depends upon shipboard conditions, such as, con-

*Senior Marine Engineer, Navy Yard, Boston.

venience for brush arm supports, interference with other shaft instruments, and accessability.

The practice prior to the emergency was to calibrate in the shop those lengths of main shafting upon which the external type electrical torsion meter would be required to be located by shipboard installation plans for the trials. From the data taken from the mechanical pointers, which were invariably located at the middle of the length of the shaft with the usual distance of 10 feet between the pointers, an average value of modulus of rigidity was determined, and at the same time readings were taken of the torsion meters each of which were fitted to the designated shaft in its shipboard location. Both the mechanical and torsion meter readings were averaged, and constants were computed for shipboard use. From the mechanical pointer data, the value of "G". the torsional modulus of rigidity or elasticity is determined for insertion in the following formula:

$$\text{"K"} = \frac{160{,}500{,}000 \text{ LB}}{G\,(R_1^4 - R_2^4)} \text{ for a hollow shaft} \qquad (1)$$

or

$$\text{"K"} = \frac{160{,}500{,}000 \text{ LB}}{G\,R^4} \text{ for a solid shaft} \qquad (2)$$

where:

"K" is the torsion meter constant to be further modified after installation by the electro-magnetic circuit adjustment, not pertinent to this discussion. (Abstract.)

"L" is the length between clamping planes of the meter on shaft. (Inches.)

"B" is the radius of the transformer pole circles. (Inches.)

"R_1" and "R_2" are the outside and inside radii, respectively, of the shaft for hollow shaft. R = outside radius for solid shaft. (Inches.)

"G" is the modulus of rigidity determined from the mechanical pointer data from shop calibration. (Pounds per square inch.)

The determination of "G" from the mechanical pointer data allowed the substitution of different torsion meters for use during ship trials through the above formula. This fact was sometimes found convenient when the torsion meters used in the shop calibration were not available for ship trials of the same vessel, but was never considered to be desirable as will be shown below.

From the data obtained from a shop calibration of stern tube shafts, the possibility of substitution of different torsion meters was found to depend upon the location of the units upon the shaft, if an error of over 3 per cent at full power is to be avoided. This same error is also likely to be encountered if care is not taken in locating the identical torsion meter used in the shop calibration on a shaft aboard the vessel, in the proper location. First, consider that some of the propulsion shafting has attached to it, by shrinking, fairly heavy sleeves; that at the end of a hollow shaft fitted with a "muff" type of coupling the inside diameter of the shaft is reduced to compensate for the taper and keyways; and near flanges and other designs of coupling there is expected to be disturbances in stress distribution from that which is encountered along the middle of a length of a shaft where the diameter is constant over a considerable length. From the basic formula

$$G = \frac{M \times X}{\phi \times I\rho} \tag{3}$$

where:

"X" is length of shaft. (Inches.)

"G" is the modulus of rigidity or elasticity of the shaft. (Pounds per square inch.)

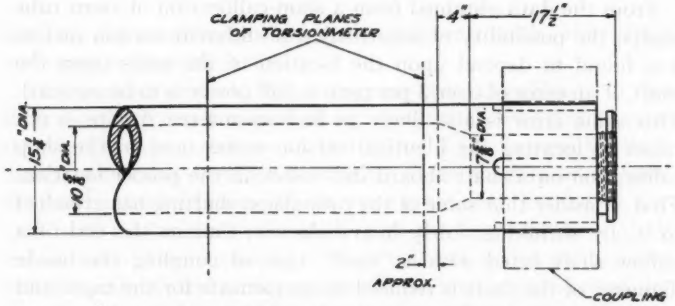
"M" is equal to twisting moment. (Pound Inches.)

"ϕ" is the angular deflection of a shaft under a twisting moment. (Radians.)

"$I\rho$" is the polar moment of inertia. (Inches^4.)

It is readily seen that a virtual increase or decrease in the moment of inertia "$I\rho$" in the above formula, caused by the factors mentioned above, would likely result in different values of "G" for an external type of torsion meter located near the end of a shaft from that obtained when the meter is mounted with several feet of clear shafting upon either side of it.

During a routine shop test of a pair of stern tube shafts mentioned above, it was decided to place an additional torsion meter between the mechanical twist measuring pointers at the center of the shafts. This meter will be called the second or No. 2 torsion meter throughout the discussion. At the same time the regular or No. 1 torsion meter was placed in its proper position as required by installation drawings, at the end of the shaft as shown by Figure 1. Each shaft was over forty-feet in



LOCATION OF REGULAR (NO. 1) TORSIONMETER ON SHAFT

FIG. 1

CALIBRATION DATA FOR TORSIONMETERS

LOAD NO.	STARBOARD SHAFT			PORT SHAFT		
	TOTAL MOMENT INCH POUNDS	CORRECTED NO. 1 TORSIONMETER READINGS	CORRECTED NO. 2 TORSIONMETER READINGS	TOTAL MOMENT INCH POUNDS	CORRECTED NO. 1 TORSIONMETER READINGS	CORRECTED NO. 2 TORSIONMETER READINGS
0	0	0	0	0	0	0
1	1,622,400	44.2	42.0	1,630,200	43.1	42.2
2	2,550,600	70.4	67.8	2,597,400	69.2	67.5
3	3,486,600	96.6	93.5	3,494,400	93.1	91.0
4	4,418,700	122.9	118.6	4,422,600	117.0	114.3
5	5,350,800	149.7	143.0	5,358,600	142.0	137.8
6	4,641,000	128.6	124.1	4,742,400	125.7	121.9
7	3,712,800	102.8	99.7	3,712,800	99.2	96.4
8	2,784,600	76.0	74.1	2,792,400	75.0	72.8
9	1,817,400	49.1	48.1	1,856,400	48.7	47.0
10	0	0	0	0	0	0

FIG. 2

DATA FOR R.P.M. – S.H.P. CURVES

R.P.M.	TORSION-METER READINGS	STARBOARD SHAFT		PORT SHAFT		TOTAL S.H.P.	
		NO. 1 $K_1 = 1.7430$	NO. 2 $K_1 = 1.6820$	NO. 1 $K_1 = 1.6741$	NO. 2 $K_1 = 1.6286$	NO. 1	NO. 2
150	13	1118	1159	1164	1197	2282	2356
175	20	2006	2080	2090	2150	4096	4230
200	24	2980	3090	3105	3195	6085	6285
225	36	4645	4810	4840	4975	9485	9785
250	44	6600	6840	6860	7060	13460	13900
275	56	8840	9150	9200	9460	18040	18610
300	66	11360	11770	11820	12170	23180	23940
325	77	14360	14880	14950	15380	29310	30260
350	87	17480	18100	18200	18700	35680	36800
375	95	20430	21160	21260	21870	41690	43030
400	105	24060	24960	25080	25800	49140	50760

FIG. 3

length. The coupling indicated in Figure 1 was used in the test, thus, simulating the shipboard installation insofar as the end connection was concerned.

Figure 2 gives the tabulation of the total moments applied to the pair of stern tube shafts together with the corrected torsion meter readings for both the No. 1 regular meter and the second, or No. 2 meter over the entire range of calibration. The differences between the corrected readings of the two torsion meters on each shaft are to be noted.

In the first two columns of Figure 3 are tabulated the Rpm. and corresponding torsion meter readings that would be expected for various speeds of a vessel fitted with the same shafting. This data was ascertained from previous trials. At the head of the next four columns in Figure 3, the torsion meter constants based on data of Figure 2 are given. These constants were obtained by inserting the data of Figure 2 in the following formula:

$$\text{Meter Constant } \text{"}K_1\text{"} = \frac{\Sigma \text{ Meter reading} \times 33{,}000}{\Sigma T \times 2\pi} \qquad (4)$$

where:

"T" is the torque producing the meter readings. (LB.FT.)

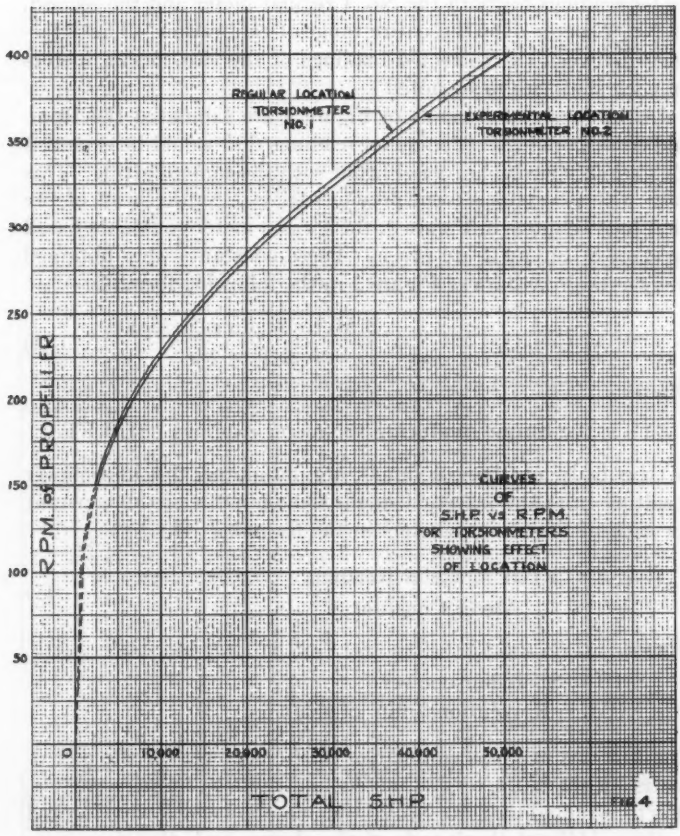
In the body of these columns are tabulated the horsepower resulting from the insertion in the denominator of the constant for each of the torsion meters in the formula

$$SHP = \frac{RPM \times \text{Meter Reading}}{\text{Meter Constant}} \qquad (5)$$

where:

SHP = Shaft horsepower
RPM = Revolutions/min. of shaft.

The last two columns summarize the horsepower for the pair of shafts for the No. 1 torsion meters and the No. 2 torsion meters. It is to be noted that the No. 2 torsion meters which were located in the middle of the length of the shafts, and, therefore were subjected to lesser movement caused by the twisting moment under tests, give greater horsepower readings than for the regular or No. 1 torsion meter located in its proper shipboard position at the end of the shaft, where there is slightly less rigidity due to the fact that the keyways are not quite compensated by the change


REGULAR LOCATION TORSIONMETER NO. 1
EXPERIMENTAL LOCATION TORSIONMETER NO. 2
CURVES OF S.H.P. vs R.P.M. FOR TORSIONMETERS SHOWING EFFECT OF LOCATION
R.P.M. of PROPELLER
TOTAL S.H.P.
400
350
300
250
200
150
100
50
0
10,000
20,000
30,000
40,000
50,000
FIG. 4

in inside diameter at this point. This is clearly indicated by the comparative torsion meter readings in Figure 2 which shows greater angular movement or twist for the No. 1 meter at the end of the shaft. Figure 4 plots the results tabulated in the last two columns of the table in Figure 3. The discrepancies in the readings of the No. 2 torsion meter in comparison with the readings of the regular, or No. 1 meter, are shown to vary by these curves, from about 2½ per cent at 200 Rpm. to 3⅓ per cent at 400 Rpm., in excess of the correct shaft horsepower values.

The following conclusions are to be drawn from the results of this experiment:

(*a*) A torsion meter must be located in the same position on the shaft on board ship for trials as it was during the shop calibration. Conversely, when calibrating the shaft ashore the meter should be in the same location required by plans for shipboard trial installations. If this procedure is not followed it is obvious from the foregoing discussion and the curves of Figure 4 that the trial horsepower readings will be subjected to error.

(*b*) Also, the former practice of substituting the value of "G," the modulus of rigidity obtained from the mechanical shop calibration, in the formula in the second paragraph will lead to similar error as described above because the mechanical data is that taken for a clear length of the shaft usually a distance of 10 feet, in the center of the length. Therefore, this value will be true only for that particular part of the shaft, and particularly in the case where the torsion meter is located near the end of the shaft.

(*c*) Prior to shop calibration, each shaft should be completely machined with keyways, sleeves, etc. Rubber covering only, need not be in place. This procedure will insure same characteristics during calibration as for actual trials.

(*d*) These tests have brought out the fact that the Electrical Torsion Meter is an accurate and sensitive instrument. All care and precaution prescribed by the manufacturer and the Bureau of Ships should be observed for installation and upkeep.

SALVAGING OF THE STEAMSHIP *WINDHUK*.

By Lewis Hobart Kenney, B. S., M. E., Civil Member.*

The German steamship, *Windhuk*, was interned in Brazil during World War II. The crew severely sabotaged the main propulsion and the electrical machinery not required for their comfort while living on board. The extent of the sabotaging was not discovered until after the crew had been removed from the ship.

The ship was a potential naval auxiliary of considerable importance and it was decided to salvage her for conversion into an auxiliary for the United States Navy, but to get the ship to the United States required the installation of sufficient propulsion equipment to maintain convoy speed.

The Philadelphia Navy Yard was designated to assemble the propulsive equipment, to prepare the installation plans, to arrange for the shipment of the propulsive equipment to Brazil and later to install it in the ship and place it in operating condition.

The *Windhuk* was a super-liner built in Germany under the extensive program set up by the Third Reich for the expansion of the German Merchant Marine. She was named for the Capitol of the former German South-West Africa, a small village, the metropolis of a sheep raising country. The ship was built by Blohm & Voss of Hamburg, and she made her first commercial voyage in 1938. The vessel was assigned to the Dutchse-Afrika Line, and her voyages were from Hamburg to Capetown, Africa. Her four cargo holds of 338,624 cubic feet were loaded on the return voyage with wool, cotton, and hides. Her 44,322 cubic feet of refrigerated spaces carried meat to Germany. On the voyage from Germany the ship carried machinery, clothing, shoes, and other manufactured goods, and her refrigerated spaces carried fish, beer, fine wines, cheeses, and all the delicacies that tied her former colonies to the Old World.

The ship was well built and gave opportunity for German engineering and inventive ability full sway. It was apparent in

*Navy Yard, Philadelphia.

looking over the ship that adequate preparation had been made for conversion to military services such as protective bulkheads, spaces for hatches through which military equipment could be passed, facilities for installation of armament, etc.

Considerable thought was given to fire risk, and the design was such as to reduce this to a minimum. A carbon dioxide system was installed with control at the navigating bridge. Her passenger accommodations were for 152 first-class, 369 tourist class, and 178 officers and crew.

To provide comfort for the voyagers, a tile swimming pool, a gymnasium, and other recreational facilities were provided; and to limit rolling. Frahm anti-rolling tanks were used.

The turbines were designed to operate at 14,200 shaft horsepower. The design speed of the ship with twin propellers was 18 knots.

In 1939 the ship was ordered to leave for a neutral port without a passenger list and only a partial cargo. She was chased by a British cruiser, but was favored by stormy weather and finally arrived at Santos, a port in Brazil.

Most of the crew remained on board, but finally only about nine or ten were left for maintenance work. They carried out an effective sabotaging program which consisted of pouring concrete into the turbines, applying a torch to the propeller shaft roller bearings, Figure 1, dry firing the boilers, Figure 2, wrecking certain of the auxiliaries and electrical generators, switchboards, controllers, etc., except the equipment required for the comfort of the crew living on board. Spare equipment and parts of machinery were wrecked beyond repair. Sugar was dumped in the oil tanks and heated.

The installation consisted of:

Two double reduction gear turbine units, each 7100 shaft horsepower.

Two Benson boilers operating at 1200 psi.

One auxiliary boiler operating at 360 psi for auxiliary machinery and port condition.

Two 550 Kw. turbo generators.

Two 380 Kw. diesel generators.

Only one of the diesel generators was left in operating condition.

On January 28, 1942, Brazil declared war on the Axis powers, and the ship came into the possession of the Brazilian Government. The transfer of the ship from Brazilian to American

cognizance was accomplished. On May 22, 1942, a survey group from the Design and Planning Sections, Philadelphia Navy Yard, with Commander J. E. Jackson, U. S. N., in charge, left by airplane for Rio de Janeiro, Brazil, to make a survey of the work to be done to install propulsion equipment sufficient to operate one propeller and thereby provide the means for transferring the ship to the United States. The party returned on June 12.

The propulsion equipment selected by the Philadelphia Yard consisted of two Diesel engines and a reduction gear. Diesel generators were installed to provide power for the operation of the blowers supplying air for scavaging and combustion and the usual group of auxiliaries for lubricating fuel oil and jacket cooling. Figures 3, 4, and 5 show the arrangement of the machinery.

The roller bearings of the main propeller shafting which were sabotaged beyond repair by the Nazis were replaced by bearings designed and manufactured at the Philadelphia Yard. A propeller was designed and manufactured by the Philadelphia Navy Yard.

The installation consisted of:

Two Fairbanks-Morse Diesel engines—ten cylinders 16 inches by 20 inches. Each engine designed for 2000 horsepower at 300 Rpm. Originally designed for a cargo vessel.

One Falk reduction gear which reduced the speed of the propeller shaft to 96 Rpm. This gear had been in operation in an industrial plant for 25 years and was about to be scrapped. Only a minor modification to the lubrication system to permit reversal was required in addition to fitting the engines to the gear.

Two Fairbanks-Morse 400 Kw. diesel generators for generation of A.C. power.

Fuel and lubricating oil systems consisting of pumps, heat exchangers, purifiers, tanks and associated equipment.

Two motor driven Allis Chalmers single stage turbo blowers 10,800 cfm at 2 pounds discharge pressure and 3550 Rpm.

Ten starting air flasks 30 inches by 96 inches.

Two Ross No. 1405 heat exchangers.

Two Ross No. 1706 lubricating oil coolers.

The Diesel generators were equipped with Ross No. 1006 oil cooler and Ross No. 805 heat exchanger.

Two Burgess Spark Arresting Snubbers for the main engines 4 feet by 9 feet.

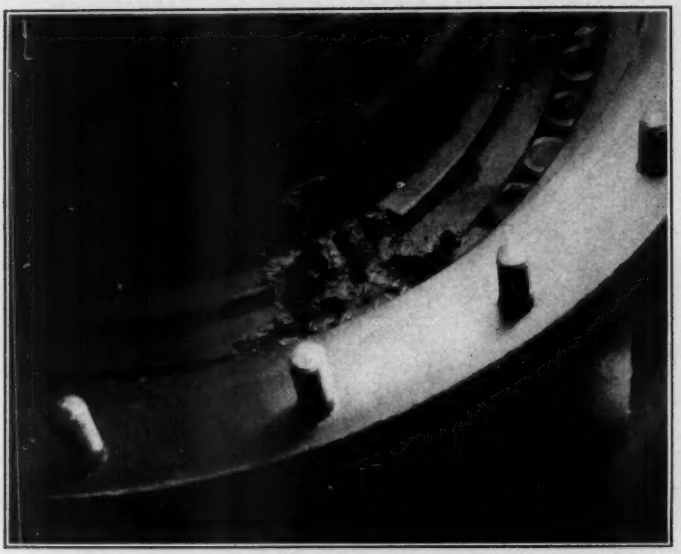


FIGURE 1.

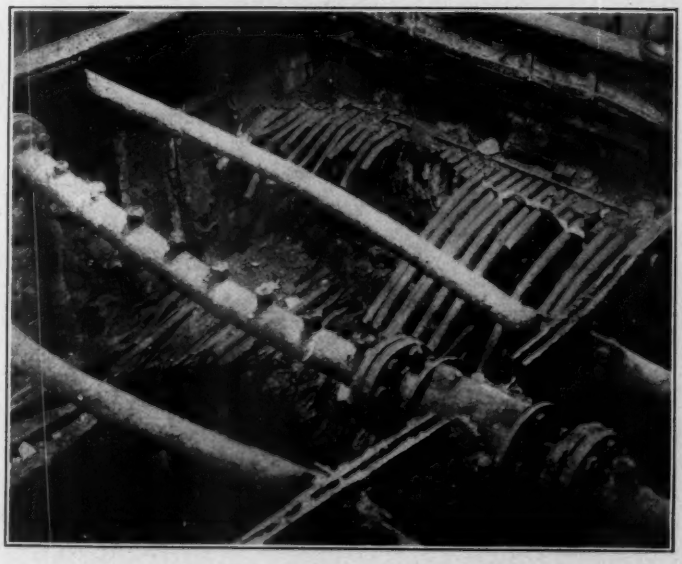
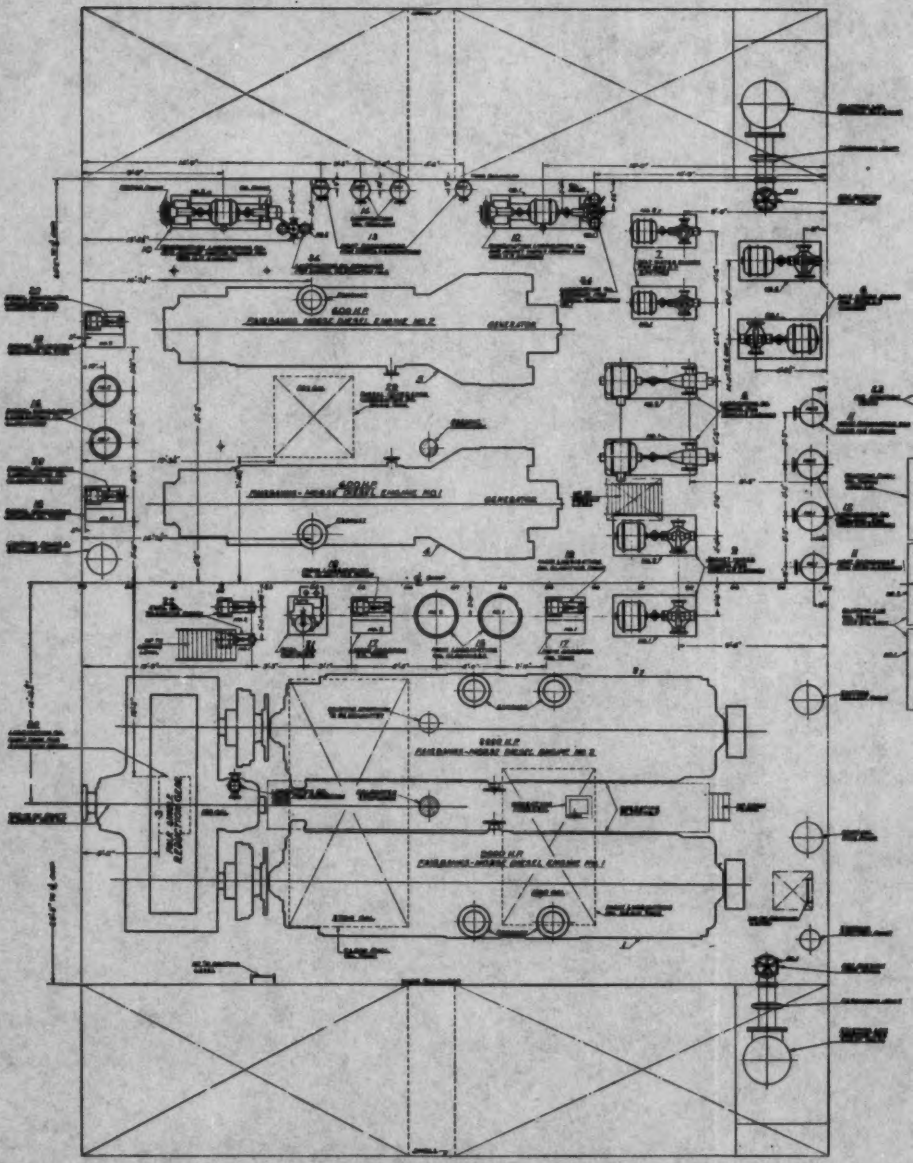
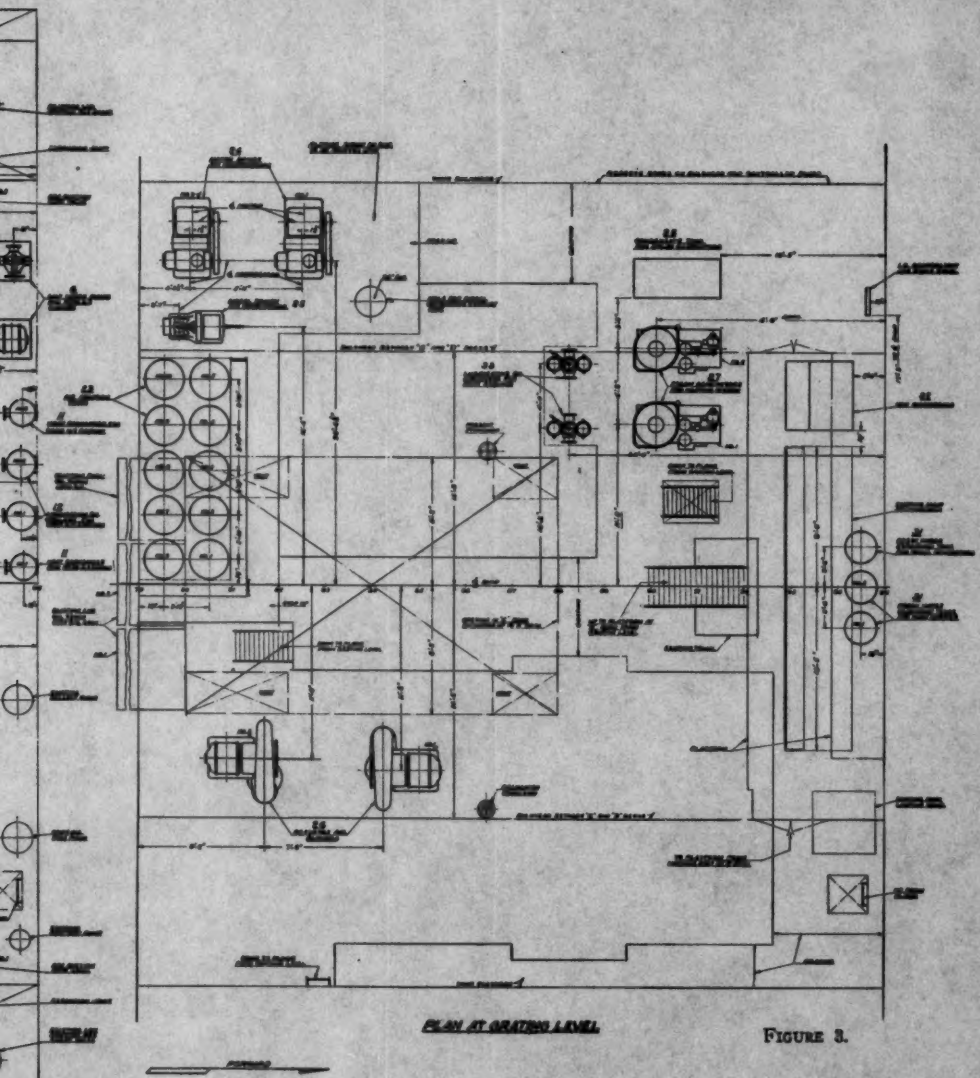


FIGURE 2.

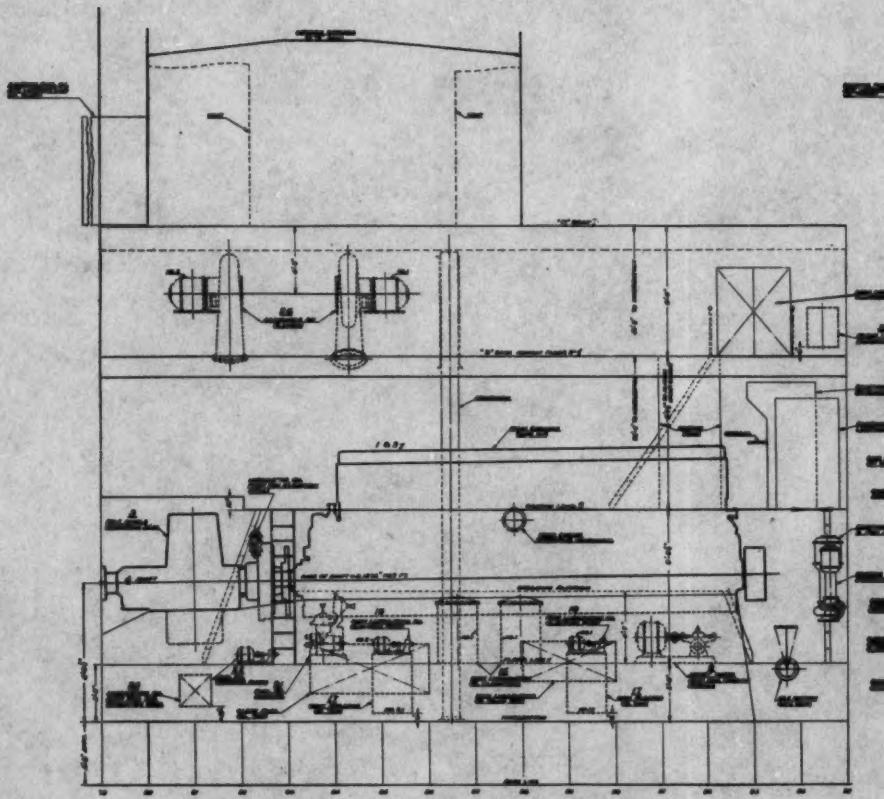


PLAN AT FLOOR LEVEL

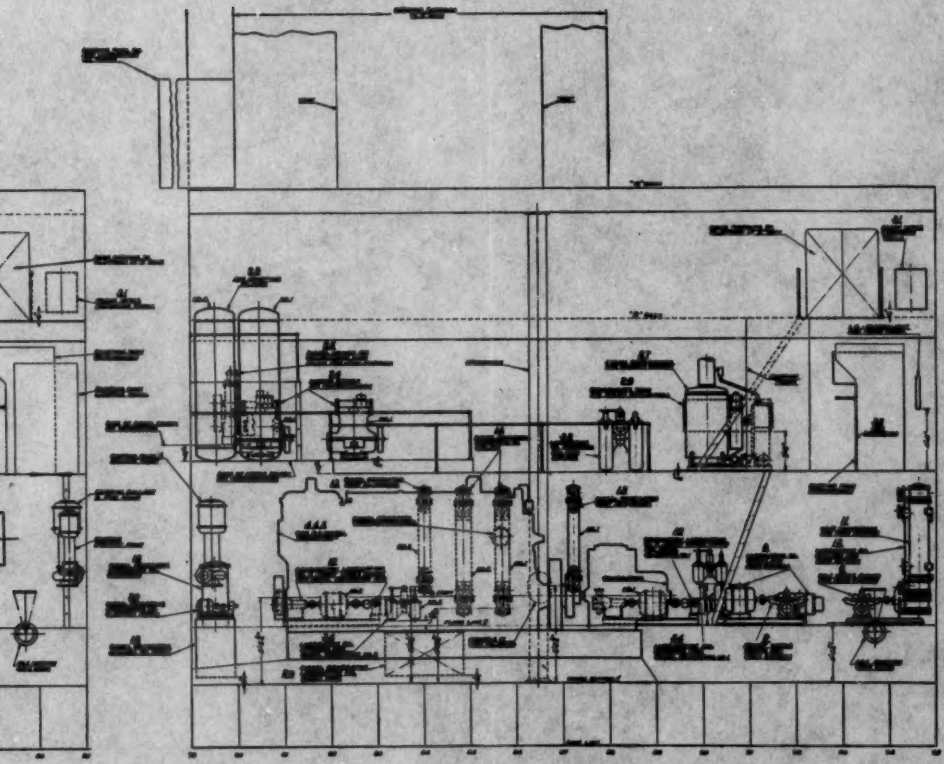


PLAN AT GRATING LEVEL

FIGURE 3.

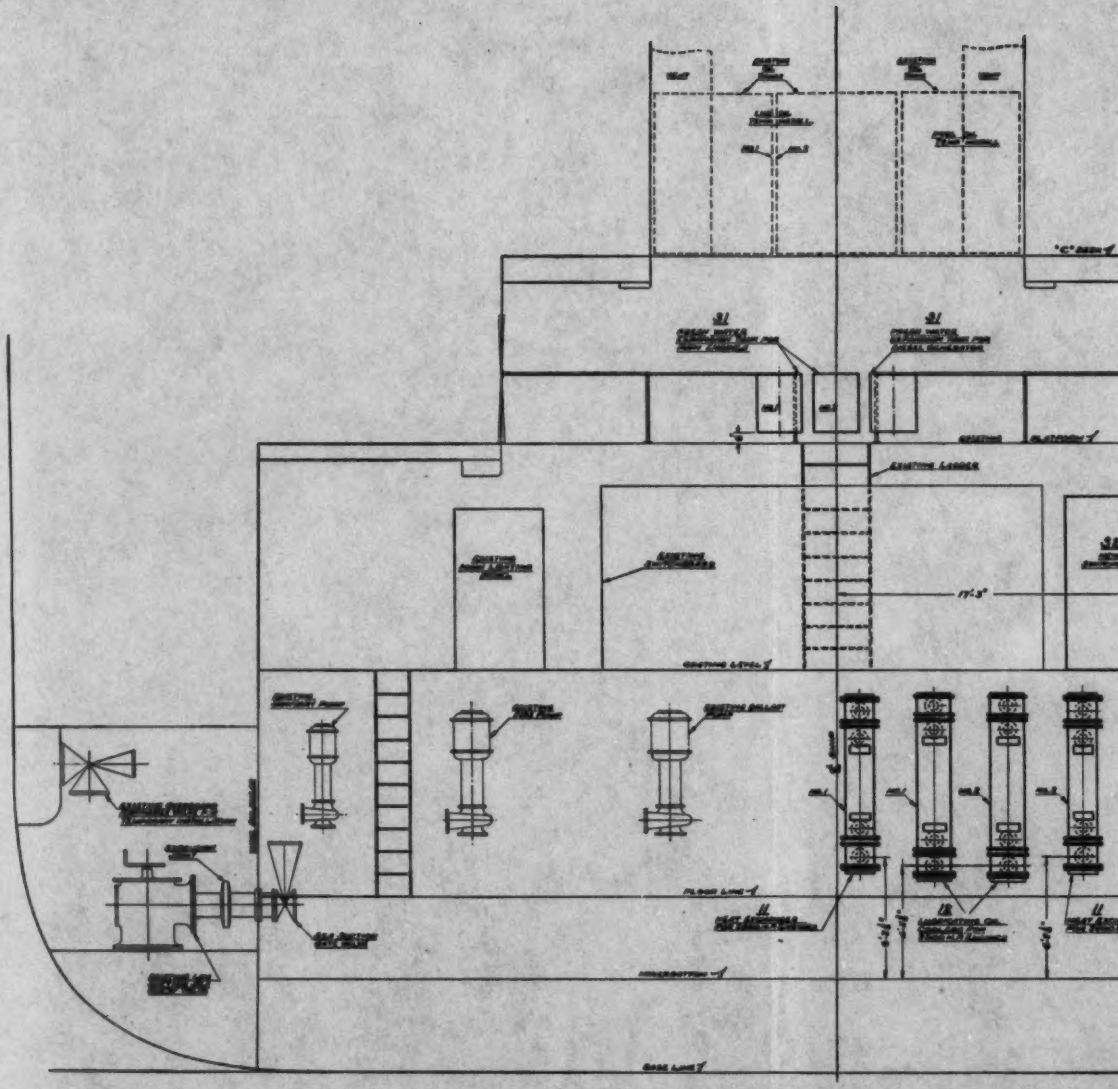


ELEVATION - STARBOARD SIDE LOOKING OUTBOARD

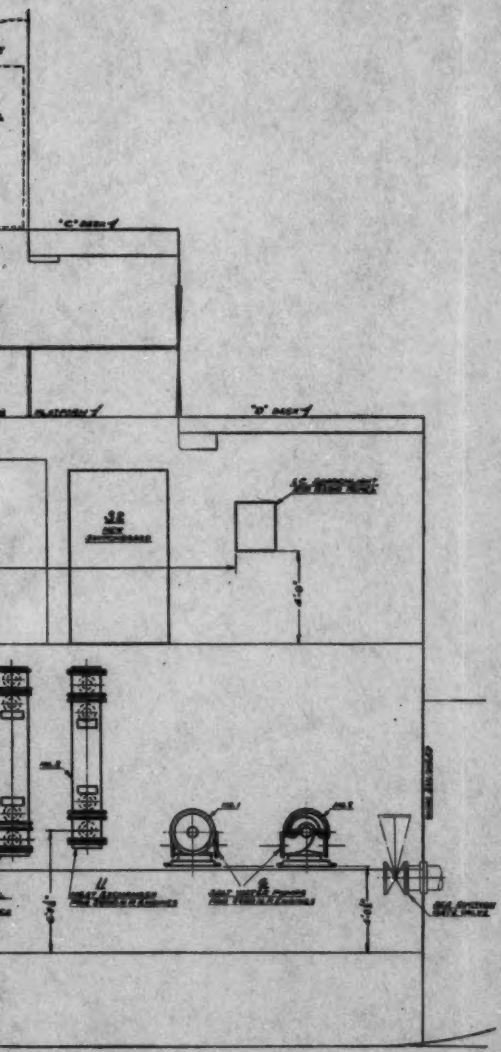


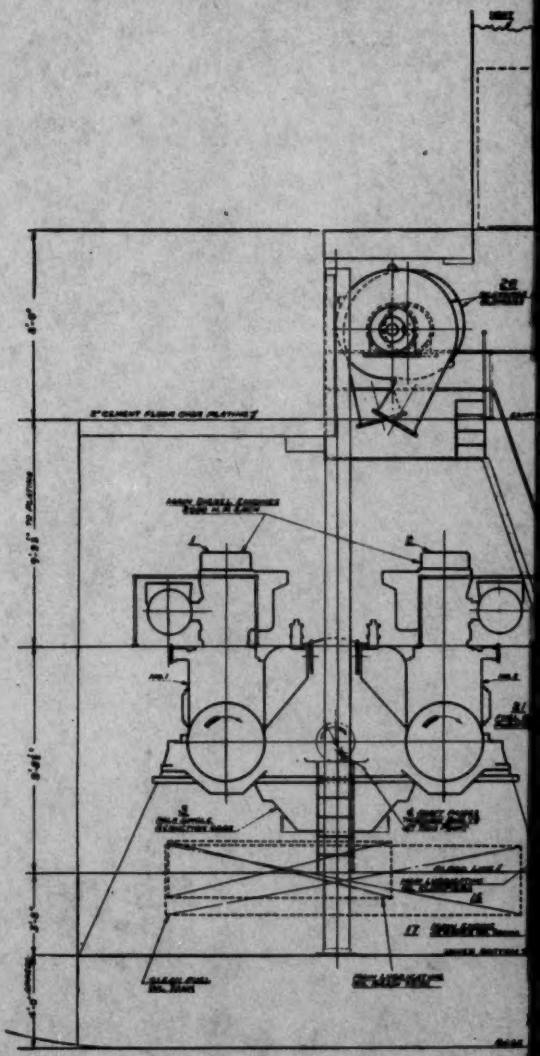
ELEVATION – CENTERLINE LOOKING TO PORT

FIGURE 4.



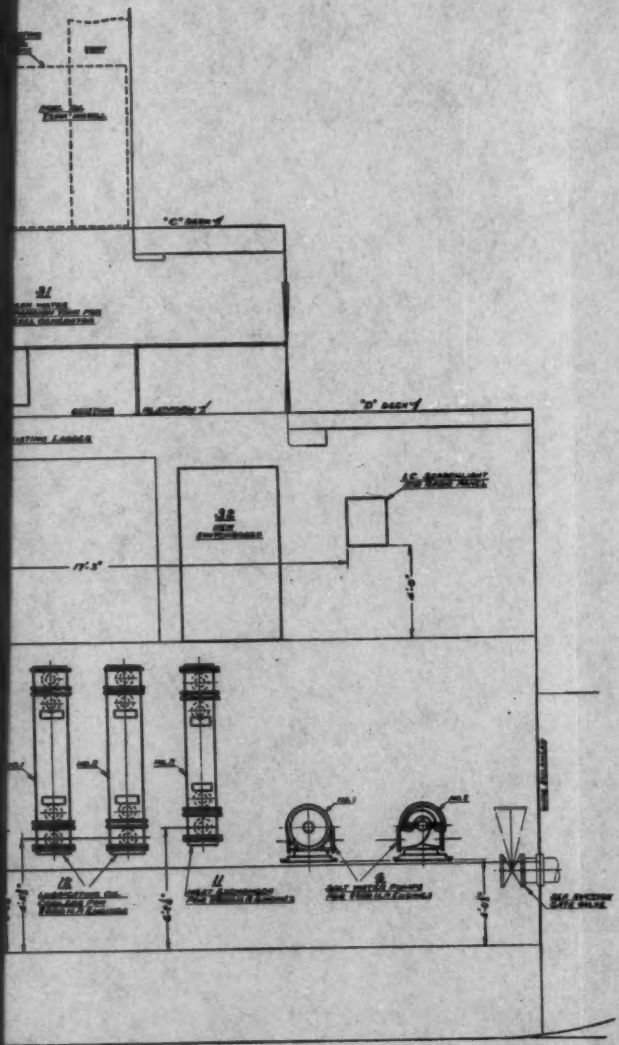
SECTION AT BULKHEAD 95 LOOKING AFT

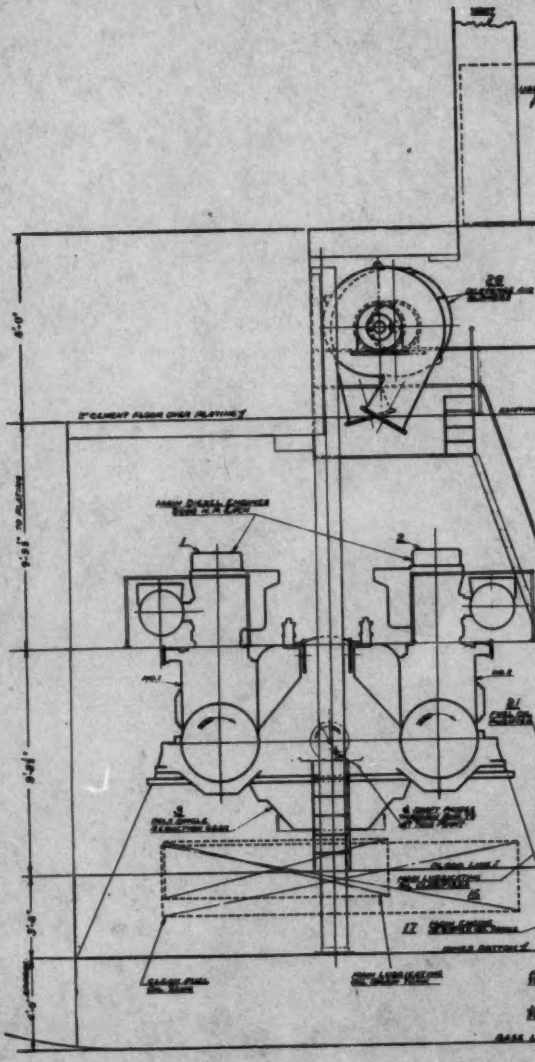




IG AFT

SECTIO





95 LOOKING AFT

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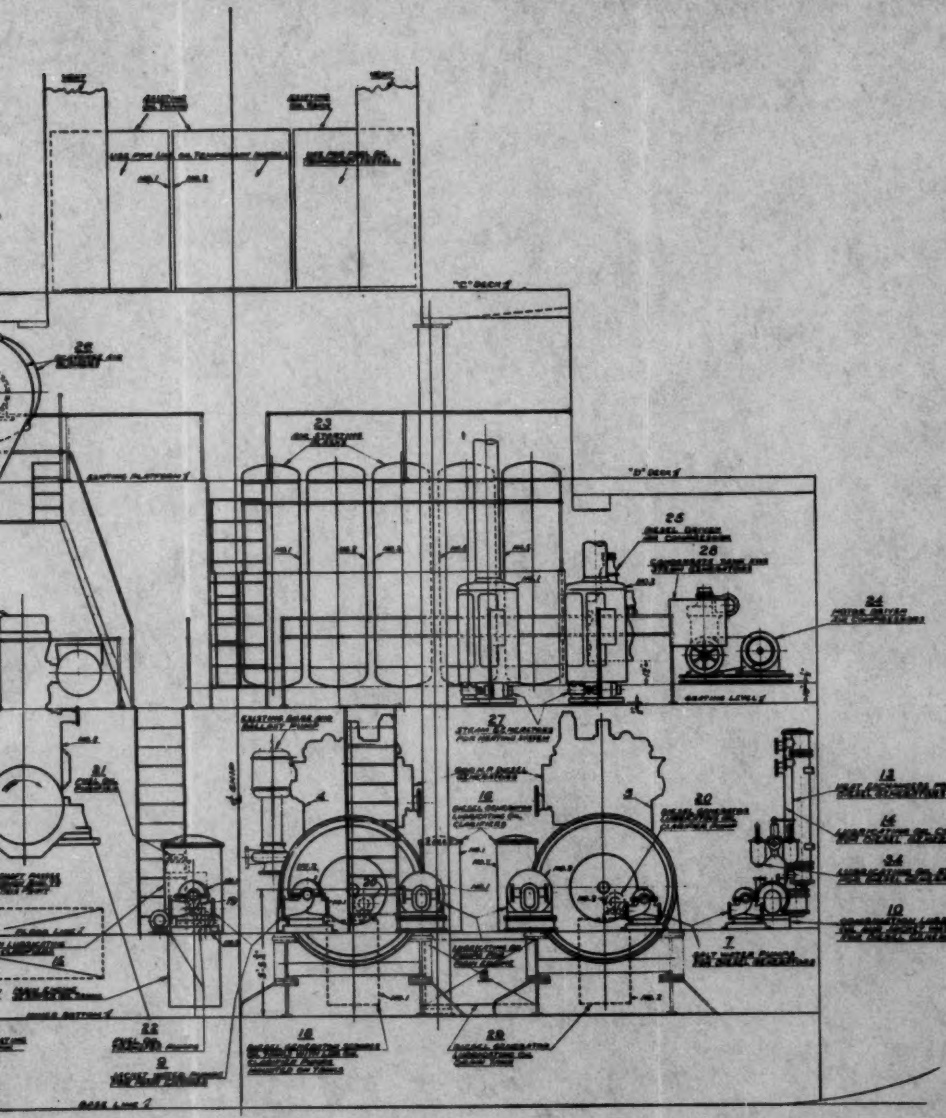


FIGURE 5.

SECTION AT FRAME 93 LOOKING AFT

Two Burgess Spark Arresting Snubbers for the generator engines 3 feet by 6 feet 6 inches.

All piping prefabricated and otherwise, valves, fittings, prefabricated foundations, required for the machinery.

Two Gardener-Denver air compressors.

One Ingersoll-Rand Diesel driven air compressor.

One propeller.

Eleven propeller shaft bearings.

One section of propeller shafting to bolt to reduction gear.

All miscellaneous equipment, such as bolts, nuts, tools, lubricating and diesel oil, and two automatic boilers for ships service.

All the equipment and material, about 650 tons, was assembled at the Philadelphia Navy Yard and shipped from there to Brazil on September 27. It arrived on November 23. The 180 men required to accomplish the installation left the Philadelphia Navy Yard and travelled by airplane to Brazil under the direction of Commander J. E. Jackson, U.S.N.

By March, 1943, the installation and the equipment had been completed, and the ship turned over to the ferry crew. On May 25 the ship arrived in the United States with 4000 tons of coffee.

In the meantime the Philadelphia Navy Yard had selected the equipment for the permanent installation. The boilers selected were of a type being built for certain naval vessels. the propulsion machinery, which consisted of turbines, reduction gears, condensers, and air ejectors, were of a type being built for certain merchant ships. The necessary plans for the installation of this equipment and auxiliary machinery were developed and forwarded to the yard designated to accomplish the installation.

New propellers adapted to requirements were designed and manufactured at the Philadelphia Yard. The generating plant consisted of three 300 Kw. turbo generators and two 500 Kw. diesel generators designed for 240/120 Volts 3 wire D.C. system.

The original electric plant of the S.S. *Windhuk* consisted of two 550 Kw., 230 volt, 2-wire, D.C. turbo-generator sets and two 380 Kw., 230 volt, 2-wire D.C. Diesel-generator sets. Emergency power was furnished by a 25 Kw., 230 volt, 2-wire D.C. Diesel-generator set.

The switchboards for the main generator were of a dead-front construction and utilized air circuit breakers for both generator and feeder protection.

The emergency switchboard was of live-front construction and provided for the control of the interior communication and battery charging circuits in addition to the emergency power circuits.

A single wire ground (hull) return distribution system was employed throughout the ship, saving considerable copper and space required for cable.

A unique feature employed throughout the ship for control of ventilation motors and for small motors in the machinery spaces was a multi-start controller. A controller of this type was installed for a group of motors, together with a group of double throw switches (one for each motor connected to panel) on a live-front switchboard-type panel. To start any motor the particular double throw switch was moved to the starting throw, thus placing the controller in the motor circuit. When the motor attained the necessary speed, a green light would burn, signalling to the operator that the double throw switch could be changed from the starting position to the running position. Upon moving this switch to the running position, the starting controller was cut out of the motor circuit and was immediately available to start the next motor. Fuses on the running side of the double throw switches provided protection from short circuits or overload.

The ship was heated electrically, utilizing recessed wall-type heaters in the staterooms, offices and other living spaces. The heaters were not of sufficient capacity to provide adequate heat in the temperate zones, but evidently were sufficient for the ship's regular run.

The *Windhuk* was equipped with excellent fire alarm and watertight door systems, engine and boiler telegraph systems, and a modern automatic telephone system in addition to a low voltage ship's service telephone system.

The steering gear system was of the Ward-Leonard type, utilizing pushbutton control at the bridge with duplicate controls in the steering gear room.

All of the ship's lighting system was 220 volt, D.C., with one contact of each fixture grounded to its mounting bracket.

The electrical installation was cleverly sabotaged. The extent of the damage to the electrical installation indicated that the Germans hoped the ship would not be taken over by the Brazilian Government for some time.

The generators were damaged by pouring concrete on the brush rigging and chopping with axes the commutator and the ends of the armature windings.

No damage was done to the switchboards, possibly because of the danger of short circuiting and damaging the remaining Diesel-generator that was capable of operation.

Throughout the ship extensive "nuisance" damage was done to ventilation motors, pump motors and controllers, and one entire steering power unit. The damage consisted of cutting the armature wiring at the commutator bars, defacing the commutators and short-circuiting contacts in the starting controllers. Practically all of the spare armatures were damaged by cutting and chopping.

The radio and gyrocompass installations were damaged beyond economical repair.

The electrical installation plans prepared by the Philadelphia Navy Yard eliminated the electrical heating system by substituting a steam heating system.

Plans were prepared for:

Power system.

Lighting system.

Interior communication system.

Other systems as required by the Bureau of Ships. Thus equipping the ship with a Navy type installation.

A survey of working conditions in Rio, indicated that skilled mechanics were unavailable and that housing conditions in the area inadequate. As the hotel accommodations on the *Windhuk* were undamaged, it was decided to berth the Philadelphia Navy Yard workmen on the ship. The only services required from the Brazilian Navy Yard were berth space, crane service, electricity, compressed air, water, etc.

It was decided, soon after the arrival of the equipment, to work two (9) hour shifts. As the work progressed, the shifts were increased to ten hours, then eleven hours, and finally two (12) hour shifts were used for about a month. In spite of the warm weather, the men were able to maintain these long shifts because no travel was involved. The walk from the stateroom to the working area was the only travel involved. Also a beer bar, serving an excellent brand of beer, was in constant use.

Each man was his own firewatch and safety man. The installation period was ninety days. The safety record was

amazingly good. No serious accidents occurred, in fact, the most serious accident was in the case of one man losing the end of his finger by means of a stateroom door. The crane operators were Brazilians, and in spite of language difficulty, the service was most excellent.

Due to the submarine activity in the South Atlantic, supplies were limited and several ingenious substitutions were made. The health of the men remained good, after usual change of climate, elements such as diarrhea, etc., were cleared up.

The writer is indebted to Comdr. J. E. Jackson, U.S.N.; Mr. J. W. Best; Mr. A. E. Henderson; and Mr. E. Cawood for information in preparing this article. They visited Brazil on this project.

NOTES.

INDEX TO NOTES.

DESIGN AND CONSTRUCTION OF WELDED MERCHANT SHIPS.
—Proceedings, Merchant Marine Council, June, 1944.

NAVAL MACHINERY: FACTORS INFLUENCING ITS DESIGN.
—Proceedings, Institute of Mechanical Engineers, Vol. 151, 1944, No. 1.

THE COMBUSTION GAS TURBINE.
—Marine Journal, June, 1944.

MISCELLANEA OF INTEREST TO NAVAL ENGINEERS.
The Rapidly Expanding Field of X-Rays.
—Journal, Franklin Institute, June, 1944.

DESIGN AND CONSTRUCTION OF WELDED MERCHANT SHIPS.

On 20 April, 1943, the Secretary of the Navy issued the precept for the Board to Investigate the Design and Methods of Construction of Welded Steel Merchant Vessels. Rear Admiral Harvey F. Johnson, U. S. C. G., Engineer-in-Chief of the Coast Guard was named as Chairman of the Board. Additional members named were: Rear Admiral E. L. Cochrane, U. S. N., Chief of the Bureau of Ships; Rear Admiral H. L. Vickery, U. S. N., Vice-Chairman of the U. S. Maritime Commission, and Mr. David Arnott, Vice-President and Chief Surveyor of the American Bureau of Shipping.

The Interim Report of the Board, dated 3 June, 1944, is reprinted herewith. It is followed by a portion of an article which was published in the June, 1944, issue of Proceedings of the Merchant Marine Council, entitled "The Structural Reinforcement of Liberty Ships." Detail specifications have been omitted from this reprint of the latter article.

1. The precept of the Board to Investigate the Design and Methods of Construction of Welded Steel Merchant Vessels requires an investigation and report. It was evident that ship failures which gave rise to the appointment of the Board were already of concern to the agencies represented by the Board. Some remedial measures had been taken before the Board was formed. Additional remedies have been ordered since that time. This report will include mention of corrective action taken, along with the results of investigation and the Board's opinion.

2. The use of welding in lieu of riveting in ship construction has a considerable history. It had an immediate appeal to the designer of naval vessels because it offered a direct means for saving hull weight. Elimination of plating and structure represented by seam laps, butt straps and faying flanges, necessary for rivets, was obvious and direct. On a given ship displacement, this saved weight could be devoted to battery, armor or other military load. The limitations on displacement imposed by the Washington and London Naval Treaties became impelling incentives to make use of light materials and welding to save weight so that a larger proportion of displacement could go into military load. The superior behavior of welded structure in way of explosion forces also became apparent. The force of these advantages made it natural for the Navy to lead the way in the application and development of welding tools, materials and procedures as applied to ships.

3. Although the advancement of the use of welding from the lesser to the more important elements of naval ship structure could not be scheduled ahead of the technical advances in welding, progress from the early '20's up to the present time has been fairly uniform, necessarily conservative and entirely free from disastrous failure. Large combatant ships still contain a considerable number of riveted connections in carefully selected joints and the end of riveting is not yet in sight.

4. In the field of merchant ship construction there is an economic incentive to produce ships with lighter hulls, made possible by welding, because the displacement devoted to cargo can be increased. The additional advantages of tightness of structure and lower maintenance cost are also realized. The character of marine risks has created a powerful, conservative force against change. Until about 1936 the application of welding to merchant ship construction was almost entirely limited to small ships and barges. From 1936 to the beginning of the war the welding of large ships was not widespread. It is interesting to note, however, that even before 1936 a few pioneers in welded merchant ship construction advanced much farther than the Navy in capitalizing the advantages afforded by welding. These advances were in the direction of shop and slab welding of large sub-assemblies and in the use of automatic welding.

5. War expansion of shipbuilding facilities brought into existence many completely new yards. It was accepted that welding art, design and performance had developed to the point where completely welded ships were practical

and fast to build. Equipping the new yards primarily for welding and with reduced fabricating tools, air compressors, piping systems, rubber air hose and pneumatic tools required for riveting operations was an important factor in the relief of manpower, toolmaking capacity and in time. These considerations influenced the decision to accept the welded hull for large mecrhant ships.

6. The classification societies and the Bureau of Marine Inspection and Navigation reached the conclusion that experience already gained with welded hulls was sufficient to warrant this extension of the practice without undue risk. Experience has shown that the war program of merchant ship construction could not have been accomplished on the necessarily large scale if the advantages of welding had not been utilized to the fullest extent.

7. At the same time, the Navy realized that, as the volume of ships and craft of a type to be built increased from the peacetime few to wartime hundreds, the usual practices of shipbuilding would have to give way to line production methods. Complete welding was the natural answer to this demand. The resulting all-welded vessels of the landing craft and destroyer escort programs paralleled the small all-welded merchant vessels in having outstandingly successful performance.

8. In contrast with the performance of small naval and merchant craft, the larger merchant vessels have experienced, in certain cases, some structural difficulties resulting in fractures in the plating.

9. The scope of the investigation necessary to survey the possible defects which have led to the fracture of ship structure afloat is very large. It involves:

A. Historical Study of Recent Hull Fractures.
   1. Statistical analysis.
   2. Specific cases.

B. Design.
   1. General.
   2. Detail.
   3. Full scale tests.

C. Materials.
   1. Base metal.
   2. Welding electrodes.

D. Construction.
   1. Procedures (fitting and edge preparation—welding techniques and welding and erection sequences).
   2. Supervisors, inspectors and workers.

E. Operating Conditions.
   1. Loading.
   2. Ballasting.
   3. Weather.
   4. Course, speed and sea routes.

F. Specific Investigations.

G. International Exchange of Information.

A. Historical Study of Recent Hull Fractures.

1. Statistical analysis.

On the basis of authenticated reports received on a representative group of 2993 ships, fractures have occurred after launching on about 577 occasions to 432 ships up to 1 April, 1944. A large proportion of these were minor in character. In the case of about 95 ships the fractures were potentially serious in that they extended well into the ship girder. Twenty vessels have suffered complete fractures of the strength deck and, of these twenty, five have completely broken in two. Two of these complete fractures occurred on vessels still at the builders' yards prior to the vessel's being placed in service. Both

of these vessels and one of the others which suffered a complete break after being in service have been repaired. Four of the vessels which suffered complete fractures of the strength deck but which had not been completely severed were abandoned so that only six vessels can be said to have been definitely lost to service from structural failures.

No lives have been lost as the result of structural failures, except in the case of the *John P. Gaines* where ten persons are missing after successfully embarking in a lifeboat.

Cracking in ships afloat has usually been associated with near freezing temperatures, or heavy seas, or combinations of these two conditions.

New fractures continue to develop but statistical data for ships afloat are insufficient as yet to indicate definitely the effect of corrective measures which have been taken. There is other evidence, however, that the incidence of fractures in new vessels while on the building ways has been materially reduced.

The statistical analysis shows that practically all fractures originate in discontinuities occasioned by design details and notch effects incidental to imperfect welding.

Contrary to popular impression, hull fractures are not confined to Liberty ships but are shared by other types of vessels.

2. Specific cases.

Appended are narrative accounts of the breaking in two of four of the five vessels mentioned in paragraph "A 1" and the account of one vessel typical of the fifteen ships which suffered complete fracture of the strength deck.

B. Design.

1. General.

The scantlings of all affected classes of vessels have been reexamined and are found to have a margin of strength over the requirements of internationally accepted standards.

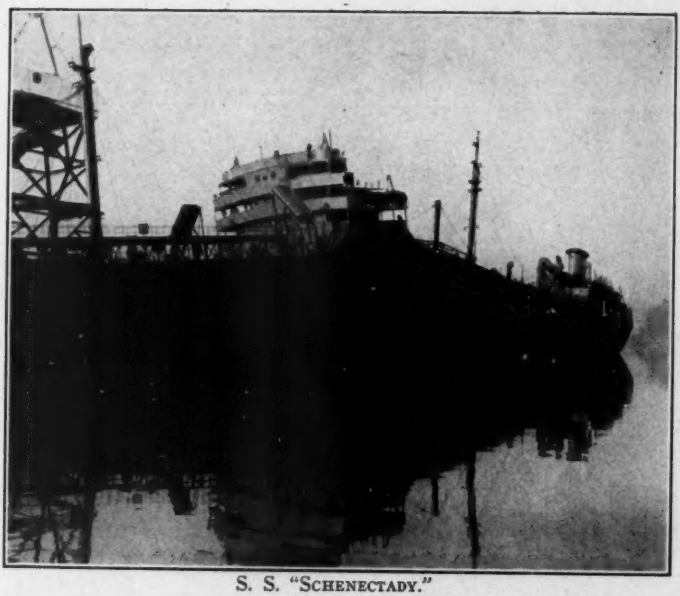
For service which involves many long voyages in ballast, the general design of future cargo ships should include provision for adequate amounts and proper distribution of liquid ballast.

It is generally accepted that the behavior of structure in the immediate vicinity of a welded joint will differ from that in way of a riveted joint because of the presence of rivet holes and the possibility of slippage in riveted joints. The configuration of riveted joints is usually different from welded joints and this may be a further contributing factor. The degree of importance of these differences in structural behavior is under investigation, with particular emphasis on stress strain characteristics of various welded joints.

2. Detail.

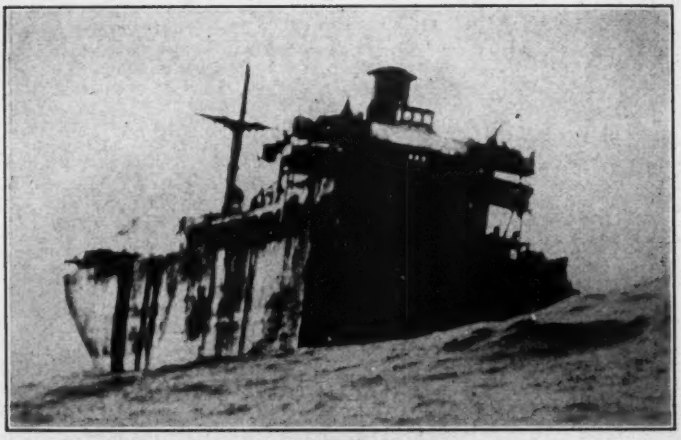
The evidence so far revealed has shown that attention to detail design to minimize notch effects of square corners and other discontinuities which has been recognized as good practice in riveted ship design, is of even greater importance in the design of welded ships. Improvements in this direction are current in unfinished construction and in all new ship designs. Directives have been issued and progress is being made toward their accomplishment in ships in service. Research is in hand at government and university laboratories to explore the character of notch effect in order to guide the continued refinement of design.

Pending the accumulation of greater knowledge resulting from service experience and research as to the origin and propagation of cracks, careful consideration should be given in new welded ship designs to the provision of means for arresting the progression of cracks once started. In the case of the Liberty ships, directives have been issued for the installation of four longitudinal riveted joints in the strength deck through the mid-length for this purpose. Work is progressing as rapidly as military considerations permit.

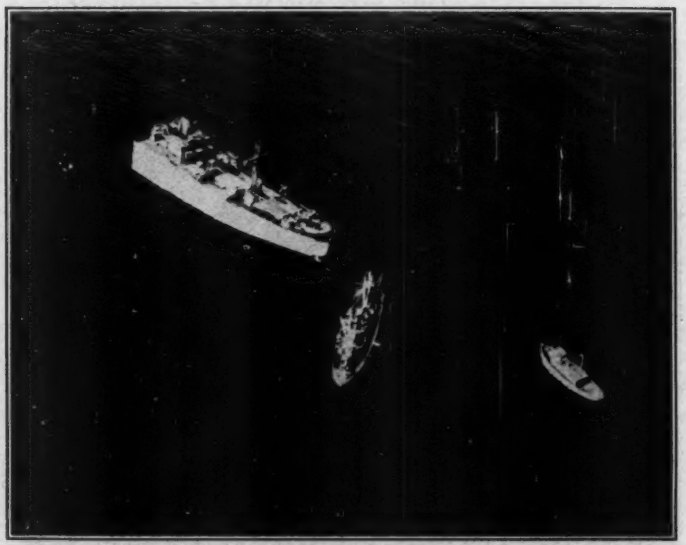


S. S. "SCHENECTADY."

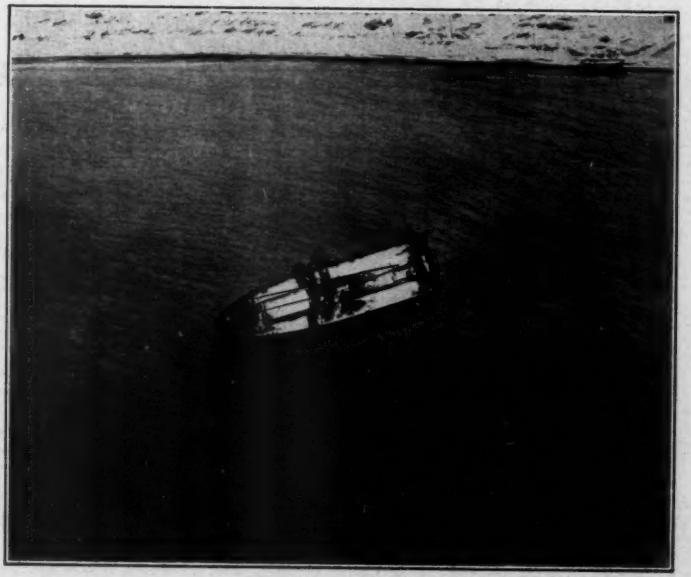
S. S. "ESSO MANHATTAN."



S. S. "JOHN P. GAINES."



S. S. "VALERY CHKALOV."



S. S. "VALERY CHKALOV."

3. Full scale tests.

Full scale hogging and sagging tests, coupled with strain measurements, have been made on five ships, one T-2 tanker, and four L-6 ore carriers. The resulting data serve to confirm earlier tests and the generally accepted agreement between calculations with the assumptions on which they are based on the one hand and the actual ship stress on the other. A full scale hogging and sagging test is presently being undertaken on a Liberty (EC-2 cargo) ship.

C. Materials.

1. Base metal

Records show that all but an insignificant portion of the steel used in the hulls of welded merchant vessels meets existing specifications for quality. In the few cases where steel has been shown to fall outside the specification requirements, this fact has not been determined to contribute to the fractures. Investigations have been initiated to explore properties not now covered by specifications with a view to improvement of material performance.

2. Welding electrodes.

There is no evidence that fractures in ships afloat have been caused by the inadequate quality of welding electrodes. Despite the pressure of the war emergency the quality and uniformity of welding electrodes have in general been maintained and, in some cases, improved through the cooperation of the producers. Investigations looking toward further improvement are being continued.

D. Construction.

Construction methods and quality of workmanship are major factors in the success or failure of welded hull structures. Some welded ships have sustained fractures while on the building ways whereas such fractures are not experienced with riveted hulls on the ways. This is accounted for in part by specific construction practices. In some cases similarity in the character and location of fractures in ships afloat to those which occur in ships on the ways has been observed, thus indirectly indicating faulty practices as a contributing cause of fracture. In other cases fractures in ships afloat have been directly attributable to faulty practices.

Recognized good practice has been shown to reduce materially the number of fractures in ships while on the ways. The means of achieving good practice are discussed in general in the following:

1. Procedures (fitting and edge preparation—welding techniques and welding and erection sequences).

The planning and execution of procedures and sequences for welded ship construction in conformance with recognized fundamentals of good practice is essential. This planning must cover the ship as a whole as well as the details of its parts. Adequate planned procedures and sequences have been available to the shipyards. The work at the University of California (see paragraph F) is pointed toward a better understanding of the basic fundamentals of good practice and of the importance and the effects of residual, "locked-up" or "shrinkage" stresses.

At the beginning of the war emergency there was general disregard by the yards of planned procedures and sequences wherever they appeared to interfere with rapid production. Subsequent to January 1943 there has been considerable improvement in this attitude through education and recognition of the serious consequences resulting from unplanned work. Further coordination of the efforts of shipfitters and welders will contribute to additional improvements. Where production pressure makes departure from the plan mandatory a new plan for local application should be developed to avoid violation of fundamental principles. A sufficient number of competent supervisors, inspectors and workers is essential for the execution of the plan.

2. Supervisors, inspectors and workers.

The number of competent members of these groups has been quite inadequate and the situation does not show any appreciable improvement as intensive programs of recruitment and training have been largely offset by the rapid growth of the industry, the large labor turnover, the demands of Selective Service, and the necessity for high speed production in the war effort.

E. Operating Conditions.

1. Loading.

Military requirements sometimes result in a loading which is less favorable to the ship, with respect to minimizing stresses in a seaway, than normal peacetime practice. The vessels, in some cases, are loaded deeper than in ordinary times and heavy deck loads are not uncommon. Although the amount and distribution of cargo always influences ship stresses, no structural fractures can be traced directly to the character of loading as the primary cause.

2. Ballasting.

Ships generally have not been more subject to fractures when in ballast than when loaded. Certain ballast conditions, in association with relatively light draft, resulting in comparatively large hogging moments, appear, however, to produce conditions favorable to the progression of cracks once started. Of the 20 ships which have broken in half or sustained complete fracture of the strength deck, 16 were in ballast and two were loaded. The remaining two of the group of 18 which failed at the builder's dock prior to going into service were adversely ballasted although not abnormally so.

3. Weather.

As noted in paragraph "A 1" a large proportion of fractures occurred during conditions of low temperature, heavy seas or combinations of these two conditions. Fifteen of the twenty ships which suffered severe damage were in heavy seas.

4. Course, speed and sea routes.

Convoy operation necessarily limits the master in choosing course and speed in heavy seas that would favor his ship. War operations have resulted in routing large numbers of ships over sea routes through colder and rougher ocean areas than was the case in peacetime.

In summation, adverse combinations of loading or ballasting, heavy seas and low temperature are factors contributing to structural distress. Of these factors only loading and ballasting are subject to control, course and speed being subject to only limited control while in convoy. Instructions have been issued in the case of Liberty ships, where the permanently installed arrangements for liquid ballast were not the most favorable, to insure a better distribution of ballast when conditions permit.

F. Specific Investigations.

The following projects of inquiry have been initiated and are being pressed to the earliest possible conclusion:

(a) Statistical analysis of steel mill products aimed at assisting the mills to obtain improved and uniform products. This subject is headed by the Bureau of Ships and contributed to by the mills, private and government laboratories.

(b) Research to show the relationship between temperature and notch sensitivity in killed, semi-killed and rimming steels at the National Bureau of Standards.

(c) "Study of the behavior of steel under conditions of multi-axial stress and the effect on this behavior of metallographic structure and chemical composition." National Defense Research Project No. NRC-77, which is being conducted by the Armour Research Foundation under the guidance of the War Metallurgy Committee (Navy Project No. NS-307).

REPORT OF STRUCTURAL FAILURE OF INSPECTED VESSEL
UNITED STATES COAST GUARD
NAVSEC-3782

This report includes all available information up to: 1 April, 1944 (date)

DESCRIPTION OF VESSEL

NAME	OFFICIAL NO.	TYPE	M.C. DESIGN
SCHENECTADY	242620	Tank Vessel	T2-SE-A1
BUILDER	**BUILDER'S HULL NO.**		**DATE COMPLETED**
Kaiser Co., Inc., Portland, Ore.	#1		31 Dec. 43
OWNER	**OPERATOR**		
War Shipping Administration	Deconhill Shipping Company		

EXTENT OF WELDING

Hull all welded
No inner bottom

Yes SIDE SHELL SEAMS			Yes DECK SEAMS
Yes SIDE SHELL BUTTS	Yes BOTTOM SEAMS	☐ INNER BOTTOM SEAMS	Yes DECK BUTTS
Yes FRAMES TO SIDE SHELL	Yes BOTTOM BUTTS	☐ INNER BOTTOM BUTTS	Yes BEAMS TO DECK
Yes BULKHEADS	Yes FLOORS TO SHELL	☐ FLOORS TO INNER BOTTOM	Yes DECK TO SHELL

CIRCUMSTANCES SURROUNDING FAILURE

(Attach all available details of ship's loading)

DATE OF FAILURE	TIME	SHIP'S LOCATION	
16 Jan., 1943	2230 P/T	Tied up at fitting out pier, Swan Island	
SHIP'S SPEED	**COURSE**	**DRAFT FWD.**	**DRAFT AFT**
0	-	6'-4"	17'-0"
SEA CONDITION	**WEATHER**	**DIRECTION OF WAVES WITH RESPECT TO SHIP**	
Still water	Clear	No waves	
WIND FORCE	**WIND DIRECTION**	**AIR TEMPERATURE**	**WATER TEMPERATURE**
Light	East wind	26° F	40° F

DESCRIPTION OF FAILURE

(Include sketch of fracture showing starting point and relative location of welds and other structural features)

APPARENT STARTING POINT: The fracture started at the juncture of the fashion plate at the aft starboard corner of the bridge superstructure and the sheer strake.

GENERAL HISTORY AND DESCRIPTION OF FAILURE, INCLUDING KNOWN CONTRIBUTORY FACTORS:

Without warning and with a report which was heard for at least a mile, the deck and sides of the vessel fractured just aft of the bridge superstructure. The fracture extended almost instanteously to the turn of the bilge port and starboard. The deck side shell, longitudinal bulkheads and bottom girders fractured. Only the bottom plating held. The vessel jack-knifed and the center portion rose so that no water entered the hull. The bow and stern settled into the silt of the river bottom. Sounding taken around the vessel eliminated the alleged possibility of the vessel having grounded amidships due to a drop in water level. A slight earth tremor was alleged to have occurred at the time of the casualty. The steel of the sheer strake was slightly below specification in yield point and ultimate strength. The deck stringer was low in yield point. Both steels were notch sensitive at low temperature, but there is no existing specification for this characteristic. The welds between the fashion plate and the sheer strake and between the sheer strake and stringer plate were found to contain defects.

CLASSIFICATION OF FAILURE: Broke in two

DISPOSITION OF VESSEL

(Repaired, lost, etc.)

Vessel repaired and put in service.

(Signed (Com. and Title))

See reverse side for loading details.

3844-27

LOADING AT TIME OF FAILURE

istribution of Weights

Forepeak	314	Long tons
Cargo tanks	0	
F.W. Tanks	71	
F.O. Bunkers Fwd.	745	On the basis of the loading
F.O. Bunkers in E.R.	486	indicated to the left, bending
I.B. Tank 11-27 P. & S.	73	moment calculations were made.
I.B. Tank 27-44 P. & S.	166	The uniform calculated stress
Ammunition Ford.	20	in the crown of the deck in
Ammunition Aft	10	still water is 10,700 lbs./in.^{2}
Dist.Water Tank	36	
F.W. Tank Aft	29	
Aft Peak	56	
Lightship	5202	
Stores & Complement	40	
Displacement	7230	Long tons
Corresponding Keel Drafts	7.0' Frd.	
	15.2' Aft	

REPORT OF STRUCTURAL FAILURE OF INSPECTED VESSEL
UNITED STATES COAST GUARD
NAVSA-2782

This report includes all available information up to: 1 April, 1944 (Date)

DESCRIPTION OF VESSEL

NAME	OFFICIAL NO.	TYPE	M.C. DESIGN
ESSO MANHATTAN	242157	Tank Vessel	T2-SE-A1
BUILDER	**BUILDER'S HULL NO**		**DATE COMPLETED**
Sun Shipbuilding & Drydock Company	# 267		22 Aug. 42
OWNER	**OPERATOR**		
Standard Oil Co. of New Jersey	Standard Oil Co. of New Jersey		

EXTENT OF WELDING

Hull all welded
No inner bottom

Yes SIDE SHELL SEAMS			Yes DECK SEAMS
Yes SIDE SHELL BUTTS	Yes BOTTOM SEAMS	☐ INNER BOTTOM SEAMS	Yes DECK BUTTS
Yes FRAMES TO SIDE SHELL	Yes BOTTOM BUTTS	☐ INNER BOTTOM BUTTS	Yes BEAMS TO DECK
Yes BULKHEADS	Yes FLOORS TO SHELL	☐ FLOORS TO INNER BOTTOM	Yes DECK TO SHELL

CIRCUMSTANCES SURROUNDING FAILURE

(Attach all available details of ship's loading)

DATE OF FAILURE	TIME	SHIP'S LOCATION	
29 March, 1943	1205 EWT	40 fathoms of water 3/4 mile inshore buoy 3, Ambrose Channel, N.Y.	
SHIP'S SPEED	**COURSE**	**DRAFT FWD**	**DRAFT AFT**
14 knots	121° True	12'-1"	18'-7"
SEA CONDITION	**WEATHER**	**DIRECTION OF WAVES WITH RESPECT TO SHIP**	
Slight ground swell	Clear	On port bow	
WIND FORCE	**WIND DIRECTION**	**AIR TEMPERATURE**	**WATER TEMPERATURE**
Force 2	Northeast	30° to 40°	Not known

DESCRIPTION OF FAILURE

(Include sketch of fracture showing starting point and relative location of welds and other structural features)

APPARENT STARTING POINT: The fracture started in a butt weld between plates A-9 and A-10 at the crown of the deck.

GENERAL HISTORY AND DESCRIPTION OF FAILURE, INCLUDING KNOWN CONTRIBUTORY FACTORS:

With a sound described variously as a thump, thud, bang, crash or explosion, the fracture ran across the deck in way of #6 tank, and down both sides, progressing to the bilge port and starboard. The vessel jack-knifed and the bow dug under an oncoming wave. The crew abandoned in the boats and were picked up by the USCG KIMBALL. The bottom fractured later and the two portions drifted apart. Subsequent examination and tests of the steel in the vicinity of the starting point and from each plate around the periphery of the hull near the fracture indicated that it met existing standards. The chemistry was normal for the class of steel. Impact and notch bend tests showed that much of the steel was sensitive to notches and low temperature. The butt weld in which the crack started contained oxide, slag and porous areas.

CLASSIFICATION OF FAILURE: Broke in two.

DISPOSITION OF VESSEL

Returned, lost, etc.

Repaired on drydock at Todd Erie Basin and returned to service.

FILLED (Date and Port):

See reverse side for loading details.

LOADING AT TIME OF FAILURE

Distribution of Weights

Forepeak		314 Long tons
Cargo Tank #1	P	388
	S	388
#2	P	6
	S	6
	Cl	1385
#3	P	16
	S	16
#4	P	20
	S	20
#9	P	21
	S	21
	Cl	1330
F.W. Tanks		70
F.O. Bunkers		240
F.O. Bunkers		240
I.B. Tank 11-27	P	41
I.B. Tank 27-44	P	82
I.B. Tank 11-27	S	31
I.B. Tank 27-44	S	84
Dist. Water Tank		16
F.W. Tank Aft		29
Aft Peak		56
L.O. Storage		2
Lightship		5455
Stores & Complement		65
Displacement		10344 Long tons

Corresponding Keel Drafts
18'-6-5/8" Aft
12'-1-1/8" Fwd.

At the time of the failure, the tanks were reported to be in the condition shown to the left. This was checked after the two portions had been towed into port. The vessel was taking ballast in accordance with the Navy schedule but due to damaged valves in #7 and #9 tanks which valves were to be repaired on the outbound voyage the schedule was not being strictly followed. The ends of the vessel contained proportionately more ballast. The uniform calculated stress in the crown of the deck in still water and in the condition noted is 12,300 lbs. per sq. inch.

It should be noted that the vessel drafts were otherwise reported to be 15' Fwd. and 22'-8" aft, also 17'-8" Fwd. and 23' aft. It is believed that the calculated drafts (to left) based upon an accurate deadweight determination made in February, 1943, are more nearly correct.

REPORT OF STRUCTURAL FAILURE OF INSPECTED VESSEL
UNITED STATES COAST GUARD
Form CG-2788

This report includes all available information up to: 1 April, 1944 (Date)

DESCRIPTION OF VESSEL

NAME	OFFICIAL NO.	TYPE	M.A.. DESIGN
JOHN P. GAINES	243861	Dry Cargo Vessel	EC2-S-C1
BUILDER		**BUILDER'S HULL NO.**	**DATE COMPLETED**
Oregon Shipbuilding Corporation		# 723	8 July 43
OWNER		**OPERATOR**	
War Shipping Administration		Northland Transportation Co., Inc.	

EXTENT OF WELDING

Yes SIDE SHELL SEAMS			Yes DECK SEAMS
Yes SIDE SHELL BUTTS	Yes BOTTOM SEAMS	Yes INNER BOTTOM SEAMS	Yes DECK BUTTS
Yes FRAMES TO SIDE SHELL	Yes BOTTOM BUTTS	Yes INNER BOTTOM BUTTS	Yes BEAMS TO DECK
Yes BULKHEADS	Yes FLOORS TO SHELL	Yes FLOORS TO INNER BOTTOM	Yes DECK TO SHELL

CIRCUMSTANCES SURROUNDING FAILURE.

(Attach all available details of ship's loading)

DATE OF FAILURE	TIME	SHIP'S LOCATION	
24 Nov., 1943	0241	#10 time zone 40 miles bearing 175° true from Chirkoff Is.	55-07 N 155-30W
SHIP'S SPEED	**COURSE**	**DRAFT FWD.**	**DRAFT AFT**
9 knots	Dutch Harbor to Seattle, 76° true	13'-0"	10'-0"
SEA CONDITION	**WEATHER**	**DIRECTION OF WAVES WITH RESPECT TO SHIP**	
Long ground swell	Fairly clear	15°-20° off port bow	
WIND FORCE	**WIND DIRECTION**	**AIR TEMPERATURE**	**WATER TEMPERATURE**
5-6 Beaufort	ENE	40°-45° F	About 40° F

DESCRIPTION OF FAILURE

(Include sketch of fracture showing starting point and relative location of welds and other structural features)

APPARENT STARTING POINT

Near Fwd. corners #3 hatch between Frs. #74 and #75.

GENERAL HISTORY AND DESCRIPTION OF FAILURE, INCLUDING KNOWN CONTRIBUTORY FACTORS:

At about 2200 on 23 November, 1943, loud noises were heard but the source could not be located in the dark. At about 0241 on 24 Nov., 1943, an exceptional sea known locally as a "freak" or "sneaker" struck the port bow, curled around the stem, and boarded near the forward gun. The fracture which had apparently commenced during the night immediately propagated. It appears that the vessel broke partially as it passed either over or between swells and the following swell completely broke off the forward end. All crew and passengers were on the after end. Survivors were picked up by U. S. Army Transports except for 10 men, including six soldiers in one lifeboat, which was lost.

CLASSIFICATION OF FAILURE: Broke in two

DISPOSITION OF VESSEL

Returned, lost, etc.

The bow is believed to have sunk. The stern is aground on Big Koniuji Island.

SIGNED (Name and Title)

See reverse side for loading details.

LOADING AT TIME OF FAILURE

Distribution of Weights

Lightship	3670	Long tons
Crew, passengers & stores	60	
Fuel oil, I.B. #1 Cl	132	
#2 P & S	313	
#3 P & S	232	
#5 P & S	232	
#6	108	
#3 deeps	197	
Settling tanks	83	
Fresh water		
Potable water tank	46	
Reserve feed tank	132	
Forepeak	115	
Aft peak	155	
Lazarette	99	
Salt water ballast		
#1 deep tanks P & S	228	
#2 deep tanks P & S	420	
Hold #4	400	
Hold #5	330	
Cargo Hold #1 Empty drums	81	
#3 Empty & full drums	442	
#4 Empty drums	47	
#5 Empty drums	50	
Displacement	7572	Long tons
Corresponding Keel Drafts	12.15'	Fwd.
	20.17'	Aft

On the basis of the loading shown to the left, indicating the condition at the time of departure from Dutch Harbor, bending moment calculations were made. The uniform calculated stress in the crown of the upper deck when on a standard wave with crest amidships is 15,600 lbs./in.^{2}

REPORT OF STRUCTURAL FAILURE OF INSPECTED VESSEL
UNITED STATES COAST GUARD
NAVCO-3783

This report includes all available information up to:

DESCRIPTION OF VESSEL — 1 April, 1944 (Date)

NAME	OFFICIAL NO.	TYPE	M.C. DESIGN
VALERY CHKALOV	None	Dry Cargo Vessel	ZC2-S-C1
BUILDER: Permanente Metals Corporation Richmond Shipyard #2		BUILDER'S HULL NO. #481	DATE COMPLETED 17 Apr. 43
OWNER: War Shipping Administration		OPERATOR: Union of Soviet Socialist Republics	

EXTENT OF WELDING

Yes SIDE SHELL SEAMS			Yes DECK SEAMS
Yes SIDE SHELL BUTTS	Yes BOTTOM SEAMS	Yes INNER BOTTOM SEAMS	Yes DECK BUTTS
Yes FRAMES TO SIDE SHELL	Yes BOTTOM BUTTS	Yes INNER BOTTOM BUTTS	Yes BEAMS TO DECK
Yes BULKHEADS	Yes FLOORS TO SHELL	Yes FLOORS TO INNER BOTTOM	Yes DECK TO SHELL

CIRCUMSTANCES SURROUNDING FAILURE
(Attach all available details of ship's loading)

DATE OF FAILURE	TIME	SHIP'S LOCATION	
11 Dec., 1943	1210	Latitude 35° N, Longitude 168°-25' W	
SHIP'S SPEED	COURSE	DRAFT FWD.	DRAFT AFT
Cut by storm	Sovetskaya Cavan Siberia to Akutan Alaska	Not known	Not known
SEA CONDITION	WEATHER	DIRECTION OF WAVES WITH RESPECT TO SHIP	
Heavy	Heavy storm, vis. 0	Apparently a head sea	
WIND FORCE	WIND DIRECTION	AIR TEMPERATURE	WATER TEMPERATURE
6 to 8	Not known	29° - 34°	Not known

DESCRIPTION OF FAILURE
(Include sketch of fracture showing starting point and relative location of welds and other structural features)

APPARENT STARTING POINT: The cracks which finally broke the vessel started exactly in the forward corners of #3 hatch port and starboard.

GENERAL HISTORY AND DESCRIPTION OF FAILURE, INCLUDING KNOWN CONTRIBUTORY FACTORS:

The vessel departed from Sovetskaya on 1 Dec., 1943, in ballast. Gales and heavy seas were encountered after departure. At noon on 11 Dec., 43, a loud report was heard and three cracks were found, one on the port side at Fr. #74, one on stbd. side at Fr. #74; and one on the stbd. side at Fr. #76. The port side crack extended from the hatch corner across the deck and down the shell to the bilge. The stbd. crack at Fr. #74 was in the side shell from the sheer strake to the tween deck. The stbd. crack at Fr. #76 ran down the side shell from the sheer strake halfway down the tween decks. The vessel was taken in tow by the tug "Joseph Stalin" but at 2206 on 13 Dec. she broke completely in two. Both portions were taken in tow by U. S. Navy tugs and brought to anchorage. The crews did not abandon ship. Ballasting details will be made available by the USSR in the near future.

CLASSIFICATION OF FAILURE: Broke in two

DISPOSITION OF VESSEL
(Repaired, lost, etc)

Both portions at anchor in Sand Bay, Great Sitkin Island. Future undet.

SIGNED (Name and Title)

(d) "Behavior of steel under multi-axial stress and the effect of welding and temperature on this behavior." National Defense Research Council Project No. NRC-75, which is being conducted by the University of California under the guidance of the War Metallurgy Committee (Navy Project No. NS-306).

(e) "Residual stress in ship welding." National Defense Research Project No. NRC-64, which is being conducted by the University of California under the guidance of the War Metallurgy Committee (Navy Project No. NS-304).

(f) "History of residual stresses in welded ship hulls of the Liberty and oil tanker types." National Defense Research Project No. NRC-74, which is being conducted by the University of California under the guidance of the War Metallurgy Committee (Navy Project No. NS-305).

(g) Experiments in strength of welding, which are being conducted by the Illinois Institute of Technology.

(h) Photo-elastic study of various means of hatch corner reinforcement, which is being conducted by the David Taylor Model Basin.

(i) Study of various types of crack arrestor by small scale models, conducted by David Taylor Model Basin.

(j) Crack arrestor study with large models, which is being performed at the University of Illinois.

(k) Weldability of steel for hull construction, NDRC No. NS 255 at Lehigh University.

G. International Exchange of Information.

Information on structural experience is being exchanged with governments of allied nations which operate American-built welded merchant vessels. The British government has organized a permanent unit known as the "Admiralty Ship Welding Committee" to study both merchant and naval ship welding problems. Information is being exchanged with this body.

Summary.

In 2561 of the 2993 large welded merchant vessels of the total Maritime Commission program in service up to 1 April, 1944, no fractures subsequent to launching have been reported. In the remaining 432 ships, fractures which were potentially serious occurred in 95, of which 6 resulted in the loss of the vessel. Analysis of the fractures indicates the existence of phenomena in welded construction which may be of importance and to which a long and satisfactory experience of riveted construction affords no reliable guide. They include such factors as shrinkage stresses built into the hulls by the welding process, the behavior of steel at low temperatures and the stress strain characteristics inherent in the locally more rigid welded structure. A large number of research projects have been initiated and are now under way in order to determine the relative importance of these factors and with a view to determining what steps should be taken in design and in construction methods to eliminate or reduce such deleterious effects as may be found. Bacisally, the abnormal conditions of wartime shipbuilding and ship operation, construction practices largely incidental to speed of production, and structural design details, are factors which in large measure have contributed to the occurrence of fractures. Appropriate steps have been taken to improve the two latter conditions where control is feasible but such corrective measures have not been in effect long enough to be able to state definitely at this time that they will prove completely effective.

In closing this preliminary report, the Board considers it important to record its opinion that without an early and general adoption of welded construction in the merchant shipbuilding program, as well as in the naval shipbuilding program, the results in speed and volume of construction which have been accomplished would have been impossible.

1. The Structural Reinforcement of Liberty Ships.

Since the inception of the "Board to Investigate the Design and Methods of Construction of Welded Steel Merchant Vessels" numerous specific steps have been taken to control the difficulties which have occurred. There has

REPORT OF STRUCTURAL FAILURE OF INSPECTED VESSEL
UNITED STATES COAST GUARD
NAVCG-2752

This report includes all available information up to: 1 April, 1944 (Date)

DESCRIPTION OF VESSEL

NAME	OFFICIAL NO.	TYPE	M.C. DESIGN
SAMUEL DEXTER (2)	243200	Dry Cargo Vessel	EC-2-S-C1
BUILDER	**BUILDER'S HULL NO.**		**DATE COMPLETED**
Delta Shipbuilding Co., Inc.	# 42		15 Apr. 43
OWNER	**OPERATOR**		
War Shipping Administration	Waterman Steamship Agency, Ltd.		

EXTENT OF WELDING

Yes SIDE SHELL SEAMS	Hull all welded		Yes DECK SEAMS
Yes SIDE SHELL BUTTS	Yes BOTTOM SEAMS	Yes INNER BOTTOM SEAMS	Yes DECK BUTTS
Yes FRAMES TO SIDE SHELL	Yes BOTTOM BUTTS	Yes INNER BOTTOM BUTTS	Yes BEAMS TO DECK
Yes BULKHEADS	Yes FLOORS TO SHELL	Yes FLOORS TO INNER BOTTOM	Yes DECK TO SHELL

CIRCUMSTANCES SURROUNDING FAILURE

(Attach all available details of ship's loading)

DATE OF FAILURE	TIME	SHIP'S LOCATION	
21 Jan., 1944	2100	Lat. 54°-48' N; Long. 22°-45' W	
SHIP'S SPEED	**COURSE**	**DRAFT FWD.**	**DRAFT AFT**
Hove to 47 RPM	United Kingdom to New York	9'-8"	21'
SEA CONDITION	**WEATHER**	**DIRECTION OF WAVES WITH RESPECT TO SHIP**	
High seas	Bad weather	3 points on stbd. bow	
WIND FORCE	**WIND DIRECTION**	**AIR TEMPERATURE**	**WATER TEMPERATURE**
8	WSW	40°	48°

DESCRIPTION OF FAILURE

(Include sketch of fracture showing starting point and relative location of welds and other structural features)

APPARENT STARTING POINT: Forward corners of #3 and #4 hatches port and starboard, all four cracks starting exactly in corner. Aft port corner #3 hatch 3 cracks. Starting point uncertain.

GENERAL HISTORY AND DESCRIPTION OF FAILURE, INCLUDING KNOWN CONTRIBUTORY FACTORS:

At 2100 on 21 Jan., deck cracked opposite #3 hatch and vessel was turned with stern to sea. At 2116 deck cracked at #4 hatch. A thorough examination was made on 22 Jan. and the two cracks from the Ford. corners of #3 hatch were found to extend across the deck and down the side to below the 2nd deck port and starboard. The crack across the deck from the stbd. fwd. corner of #4 ran down the side below the waterline. The top of #3 deep tank in #4 hold was partially cracked. The crack from the port fwd. corner of #4 ran across the deck a distance of 2'. The weather moderated during the 22 to 24 January but a watch was kept on the cracks which were gradually increasing and opened and closed 1" in the seaway. Bad weather was forecast so between 1530 and 1630 on 24 January, the vessel was abandoned.

CLASSIFICATION OF FAILURE: Cracked deck.

DISPOSITION OF VESSEL

Operated, lost, etc.

Vessel drifted ashore on Barra Island of the Hebrides. Future undet.

SIGNED (Name and Title)

See reverse side for loading details.

464921

LOADING AT TIME OF FAILURE

Distribution of Weights

Lightship	3670	Long tons
Crew, passengers & stores	30	
Fuel oil, settling tanks	100	
Deep tank #3	613	
Fresh water, potable water tank	45	
Reserve feed	132	
Forepeak	141	
Aftpeak	155	
Salt water ballast		
Deep tank #2	420	
Inner bottom #1	144	
#2	559	
#3	252	
#5	252	
#6	118	
Ballast in holds #2	250	
#3	250	
#4	500	
#5	500	
Displacement	7911	Long tons
Corresponding Keel Drafts	10.51	Ford.
	23.05	Aft

On the basis of the loading shown to the left, bending moment calculations were made. The uniform calculated stress in the crown of the upper deck when on a standard wave with crest amidships is 16,700 lbs./in.^{2}

been a great deal of conjecture in shipbuilding and shipping circles regarding the measures which have been taken and their intended purpose. It has been impossible in the past to disclose the problems in their true light, but now that the difficulties are more fully understood and the scope and gravity of the situation can be evaluated, it is possible to set down the problems that presented themselves and the steps which were taken through the combined efforts of the U. S. Navy, U. S. Coast Guard, U. S. Maritime Commission, and the American Bureau of Shipping.

The first step taken was to educate the shipyard personnel, inspectors and surveyors in the more obvious causes which were contributing to the structural failures. Three pamphlets, whose contents were quite similar, were issued. The Coast Guard and American Bureau of Shipping issued instructions to their inspectors and surveyors, and the Maritime Commission printed a handy pamphlet which was circulated to all shipyard personnel in supervisory capacities and to all inspectors and surveyors. At the same time orders were issued to the shipyards to make specific alterations in two or three items on new vessels which were known to have contributed to the failures. These included the elimination of a cut in the sheer strake of the Liberty ships in way of the gangway, freeing of the bulwark from the top of the sheer strake on the Liberty ships, serrating the bilge keel on both the Liberty ships and T-2 tankers, and adding large girders beneath the deck of the T-2 tankers.

By the end of the summer of 1943, the more obvious difficulties had been largely removed from new ships, but it was apparent that there was a tendency in the repair yards to conclude that, since a plate had cracked, it must of necessity have been rolled of defective material. The ship repair yards were removing large quantities of plating in effecting the repairs. Study showed, however, that the material complied with existing specifications and that new material introduced in the repair was, in all probability, in no way superior to that removed. Instructions were therefore issued in Marine Inspection Memorandum No. 64 covering the repairs of structural failures.

It also became evident that the majority of reports of structural failures were not complete enough to permit study of the more elusive causes. New forms were therefore devised for making the reports so that important information would not be omitted. Large diagrams were drawn for the Liberty ships and T-2 tankers on which the location of the fracture or buckling could be indicated and on which spaces were provided for all of the pertinent data so that inadvertent omission of important facts would be avoided. A third form of letter-size, NAV-CG 2752, was devised for reporting structural failures on vessels other than the Liberty ships and tankers. In order to establish definite procedures for making these reports, Marine Inspection Memorandum No. 57 was issued. All of these data were distributed, not only to the Merchant Marine Inspectors of the Coast Guard, but to the Maritime Commission, Navy Department, American Bureau of Shipping, the British Corporation, and the British Admiralty, and at the present time structural failure reports are being received through the regular channels of all these agencies.

The structural reinforcements to be added are intended to serve two purposes: First, steps are taken to eliminate as many crack starters as possible; and second, barriers are introduced to limit the progress of any crack which might start from an undetected origin.

It has been found that practically any square corner introduced into the material may prove a source of difficulty. Square corners are therefore to be rounded in so far as practicable, and, in the case of the hatch corners which are a proven source of difficulty, specific reinforcements are to be added. Figure 1 shows the original reinforcement provided on most of the existing Liberty ships.

On new vessels an improved corner reinforcement is being fitted during the construction period. The reinforcement for the new vessels is shown in Figures 2 and 3. Orders for this change were issued in August, 1943. On vessels which were not reinforced at the hatch corner, as shown in Figures 2 and 3, a rounded insert corner with face plate is being fitted as shown in Figure 4.

On the earlier Liberty ships a bulb bar bilge keel was used. In welding the butts care was seldom taken to see that the entire section was penetrated by the welding and a cavity was generally left in the heart of the bulb. The bilge keel is the outermost bottom fiber of the ship's hull girder and as the vessel bends and twists in the seaway it is severely strained. About half of the underwater shell fractures have been traced directly to a defective butt-weld in a bilge keel.

On vessels whose bilge keel had not been serrated during construction, holes are to be made opposite the butt welds, as shown in Figures 5 and 6.

Since there is a possibility that some cracks may emanate from undetected starters, such as cracked or imperfect welds, a barrier is to be provided to arrest such cracks before extensive damage results. From years of experience on riveted vessels, it is known that a riveted seam forms a barrier to a crack and, although an extensive study is still underway, no barrier has yet been found which is more satisfactory than a riveted seam. In reinforcing the Liberty ships, a riveted seam was, therefore, provided at the gunwale by cutting either the stringer plate or the sheer strake and fitting a riveted gun-

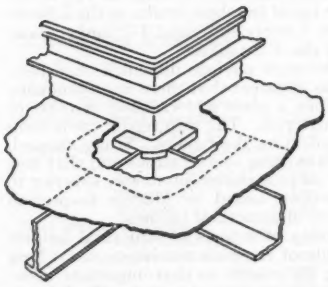


Figure 1.—Outside view of hatch corner in upper deck of earlier Liberty ship shows original design including 51 pound doubler beneath deck and welding around insert plate.

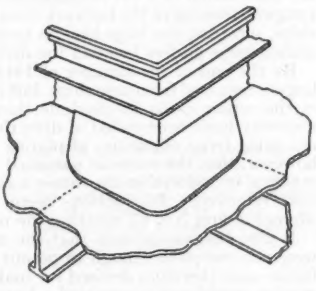


Figure 2.—Outside view of hatch corner in upper deck of new Liberty ship. The 20 pound doubler is shown on top of deck and covering insert plate.

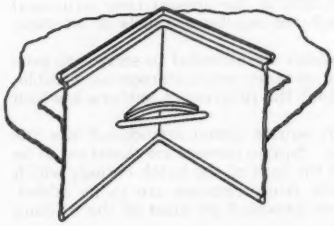


Figure 3.—Inside view of hatch corner in upper deck of new Liberty ship. The 20 pound doubler plate and insert plate are shown slotted through the hatch coaming and rounded to a one foot radius.

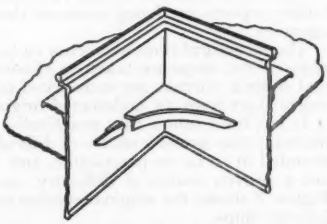


Figure 4.—Inside view of hatch corner in upper deck of Liberty ship. The corner reinforcement to be provided on existing vessels is shown in place.

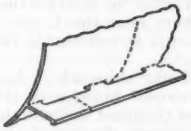


Figure 5.—Bilge keel showing serrations such as would be provided on a new vessel.

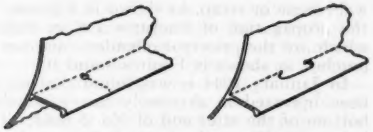


Figure 6.—Bilge keels on existing vessels showing holes provided at each bilge keel butt on dry cargo and tank vessels (left) and troop-ship conversions (right).

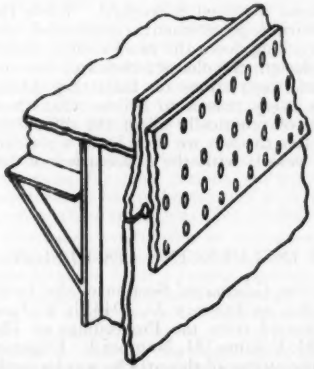


Figure 7.—Sheer strake strap showing torch cut in sheer strake just beneath upper deck.

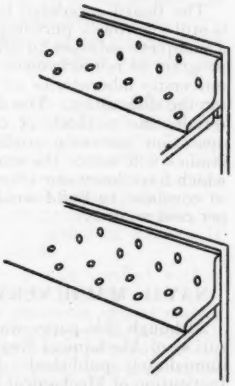


Figure 8.—Riveted gunwale angle showing arrangement for existing vessels with torch cut through stringer plate (above) and for new vessels (below).

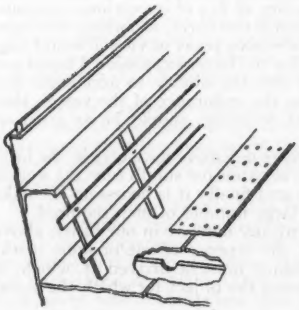


Figure 9.—Deck strap beside hatch on dry cargo or tank vessel.

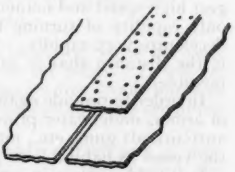


Figure 10.—Deck strap on troop-ship conversion.

wale angle or strap, as shown in Figures 7 and 8. In order to limit further the propagation of fractures and in order to stop them near the hatches, which are the principal offenders, another riveted seam is to run beside the hatches, as shown in Figures 9 and 10.

In January 1944 it was found that some of the Liberty ships which had been operated in extremely heavy weather had developed buckles in the bottom of the after end of No. 3 hold. These buckles occurred in both the bottom and inner bottom of the vessel. Not many vessels developed this buckling and since it appeared that it was not serious enough to occasion great concern, no steps were taken except in the case of the troopships where difficulties in ballasting aggravated the situation and where so many lives were at stake that it became advisable to lean over backward in providing extra precautions. In these vessels four additional longitudinal girders are to be fitted.

The Board appointed by the Secretary of the Navy to study the problem is still vigorously pursuing the causes of the fractures and will continue until a complete solution to all of the problems involved is reached. While the program of reinforcement is going on, various government, commercial, and university laboratories are engaged in tracking down the more remote causes for the difficulties. The details of the design, the characteristics of the materials, the methods of construction and restrictions on ballasting during operation are being studied. There is every reason to believe that these studies will locate the causes and cures of practically all of the difficulties which have beset our vessels and that after the war we shall be in a position to continue to build welded merchant vessels with the expectation of 100 per cent security.

NAVAL MACHINERY: FACTORS INFLUENCING ITS DESIGN.

Although this paper was read before the Graduates' Section of the Institution of Mechanical Engineers in London on January 23, 1943, it was not immediately published. It is now reprinted from the Proceedings of The Institution of Mechanical Engineers, 1944, Volume 151, Number 1. Engineer Vice-Admiral Sir George Preece, R. N., the author of this article, was formerly Engineer-in-Chief of the British Fleet. Admiral Preece is currently a Vice-President of the Institution of Mechanical Engineers.

To make an informed approach to the problems connected with naval machinery design it is first necessary to appreciate the qualities that are required of the vessel as a whole. Of these, its capacity for offensive action, represented by the number, size, and rapidity of fire of its various weapons, and the amount of ammunition for them that it can carry, is perhaps the most important. Next in importance from an offensive point of view I would suggest high speed and maneuverability, and by the latter expression I mean not only rapidity of turning under helm but also the ability to accelerate and decelerate very rapidly. At the same time the endurance of the vessel, that is the distance that it can steam without refueling, should be as great as possible.

In order to provide against attack, adequate defensive measures in the form of armor, underwater protection by means of extensive subdivision, as well as anti-aircraft guns, etc., must be provided; and finally it is necessary to make the vessel as habitable as possible for the large number of men required.

It is evident that on a given displacement any increase in one of the above desirable qualities can only be effected at the expense of one or more of the remainder—the result being that a compromise must be arrived at, which, in the opinion of the naval architect, best meets the object for which the vessel is being designed.

It is equally evident from the foregoing considerations that the machinery which moves the ship and which supplies all power required to work and fight

her must occupy the minimum of space and be of the least possible weight. In considering space, length and height are more important than width, provided that the last-named does not encroach on the space necessary for underwater protection. Length is important because of the desirability of keeping the machinery within the armored citadel, and height from the point of view of keeping it as far as possible below the water line. At the same time it must be "flexible," i. e. it must be capable of rapid alterations in power, for it is this quality which confers maneuverability on the ship. It must also be reliable, durable, accessible, economical in fuel and require the minimum number of men to operate it.

Like those which go to make up the ship as a whole, these qualities which it is desired that the machinery should possess are largely mutually conflicting.

For example, flexibility is chiefly a matter of steam control and can more easily be achieved with ample boiler power, the provision of which involves increased weight and space, and accessibilify by increased space alone.

Reliability is of great importance, since any failure not only puts out of action the machinery involved but may also destroy the whole value of the ship as a fighting unit.

Durability would be of less importance if a consistent scrapping and replacement policy could be counted upon; but in this connection it is only necessary to remark that nearly all our capital ships at the start of this war had taken part in the last one whilst the feats in this war of some of our old "V" and "W" class destroyers are well known. Apart from this, however, a certain degree of durability is essential, since our larger ships are required to be self-supporting for long periods, dockyard refits are infrequent, and ordinarily maintenance has to be effected at four to eight hours' notice for steam. It is under these conditions that accessibility is appreciated.

Extensive subdivision of machinery compartments is in conflict with a minimum number of personnel and although automatic apparatus, or controls operated from a central position, would be of assistance in this matter the prospect of their failure through enemy action must always be visualized, in which event the personnel necessary to "take over" must be ready on the spot. Actually, however, this is not a very real problem since in general the number of men in the engineering department is dictated by maintenance requirements of both the machinery and those parts of the hull for which they are responsible. Apart from other considerations time does not allow of my entering into a detailed account of what has been achieved by the engineer in reducing his demands for weight and space, but an example will suffice to show how things have moved in this direction. A few years ago certain of our capital ships were reengined and reboilered and some of the more striking improvements effected were as follows:—

Length of engine and boiler rooms reduced by 22.5 per cent.
Floor space occupied by machinery, including various auxiliaries such as electric generators, hydraulic engines, etc., reduced by 34 per cent.
Weight per shaft horsepower reduced by 50 per cent.

The endurance under comparable conditions was increased by 34 per cent at one end of the speed range and by 71 per cent at the other. It will be appreciated that the designer in this case did not have a free hand, but was limited by the fact that certain features of the original installations remained unchanged.

Some phenomenal results have been achieved recently in the way of producing extremely light machinery for very fast craft. As is well known, the weight per horsepower of similar machines increases as the horsepower diminishes; but in this instance, although the horsepower of the small vessel was only about one-fifth of that of the larger one, its machinery weight per horsepower was actually less than one-half of that of the larger vessel, in spite of the fact that the machinery of the larger ship was of the lightest type made hitherto.

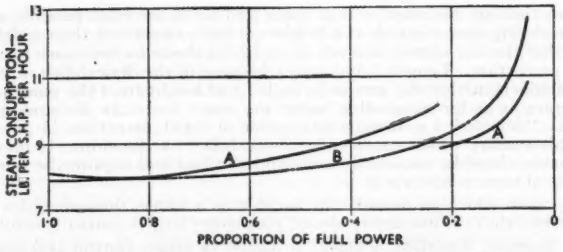


FIGURE 1.—STEAM CONSUMPTION ON A BASIS OF SHAFT HORSEPOWER.
A—All reaction high-pressure, with geared cruising turbine.
B—Impulse reaction high-pressure, with impulse wheel and two reaction stages for cruising.

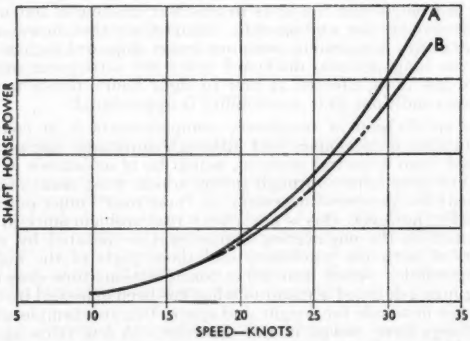


FIGURE 2.—HORSEPOWER-SPEED CURVES ON A BASIS OF SHAFT HORSEPOWER.

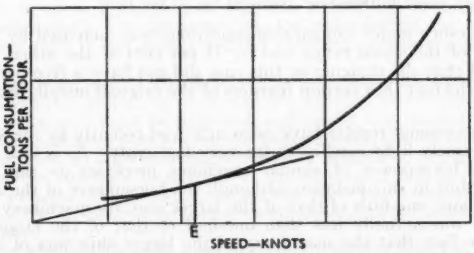


FIGURE 3.—FUEL CONSUMPTION ON A BASIS OF SPEED.

Flexibility is indicated by the following figures, which are typical of a light and heavy ship respectively:—

Stop to full ahead 45–160 seconds.
Full ahead to stop 25–50 seconds.
Full ahead to full astern 100–260 seconds.

Acceleration trials (in an escort vessel): speed rose from 9 to 21 knots in 51 seconds.

Now let us consider economy. In general it is desired that a warship should have a large endurance at war cruising speed, at which speed the power required is considerably less than that required at full speed.

Full power is required for relatively short periods, so a step in the required direction can be taken by designing the turbines so that they operate at their maximum efficiency not at full power but at a point somewhere below full power, two typical curves of steam consumption per shaft horsepower-hour (on a base of shaft horsepower) then taking the form shown in Figure 1.

Curve B illustrates this procedure, the point of minimum steam consumption per ship horsepower-hour being at approximately 80 per cent of full power. This method cannot, however, be pursued very far because it will be found that the accompanying reduced turbine efficiency at full power would soon demand unacceptably large boiler power. The situation is aggravated by the fact that the boiler efficiency characteristic is a falling one, the efficiency being a minimum under full putput conditions.

The chief difficulty in designing naval machinery to operate economically over a wide range of power is that a reduction in power is accompanied by a reduction in the speed of the ship and similarly of the turbines. The turbines therefore operate under conditions entirely different from those in an electric power station where throughout the whole power range the turbines run at a constant speed.

The propeller, and therefore the turbine revolutions per knot, remain approximately constant over a wide range of speed and at moderate speeds the horsepower varies roughly as the cube of the turbine revolutions; but at the higher powers it may vary as the fourth and even higher powers in a high-speed vessel. A typical shaft horsepower-speed curve is shown in curve A, Figure 2, whilst for comparison curve B has been drawn on the basis that the horsepower varies as the cube of the speed over the whole range. Similar curves may be plotted for fuel consumption for main turbines only, and for all purposes, on a speed base; and it will be noticed that the economical speed of the ship, i. e. that at which the maximum distance is travelled per ton of fuel expended, is obtained by drawing a tangent from the origin to the latter curve (Figure 3). A more accurate method is to plot the miles per ton and locate the peak.

In this connection it may be remarked that in capital ships the amount of steam, and consequently of fuel, used to provide power to operate its guns, etc., is a very appreciable proportion of the total, and adequate boiler power must be provided to meet this demand.

It must be remembered that the consumption speed curve varies considerably with different conditions; deep draught, dirty hull, head winds and sea,-all increase the fuel consumed at a given speed. For instance, if a ship steaming at so many revolutions per minute when just out of dock obtains a certain speed through the water and later steams at the same number of revolutions when the hull has become dirty, not only will her speed have diminished, but the horsepower required, and therefore the fuel consumption, will also have increased. Provided, however, that the ship is operating in the range in which the horsepower P varies as the cube of the speed V, and that the two conditions denoted by the suffixes 1 and 2 are not widely dissimilar, then

$$V_1 \sqrt{P_1} = V_2 \sqrt{P_2}$$

This relationship is, I think, little known but has been found to be extremely valuable as an aid to navigation; since, if torsionmeters are fitted, the speed of the ship through the water can be calculated from the power developed

under strange conditions—provided, of course, that an accurate horsepower-speed curve obtained under any conditions (so long as they are constant) is available.

As already indicated, better economy at lower powers can be obtained by designing the turbines so that their point of maximum efficiency is below full power; but this is not sufficient. Warships have to operate at all kinds of speeds for long periods, as for example a destroyer protecting a slow convoy. The turbine blade speed is much reduced whilst the potential heat drop between boiler and condenser remains approximately unchanged. A certain measure of economy can be maintained by fitting an impulse wheel as the first stage of the high-pressure turbine and arranging a number of separate groups of nozzles each with its own shut-off valve. With this arrangement it will be appreciated that the moving blades are subjected to a regular series of impulses and before the phenomena of disk and blade vibration were appreciated, and could be guarded against, cases of both kinds of vibration occurred. The problem was the more difficult with naval turbines because they were required to run for considerable periods at any desired speed and not only at one speed.

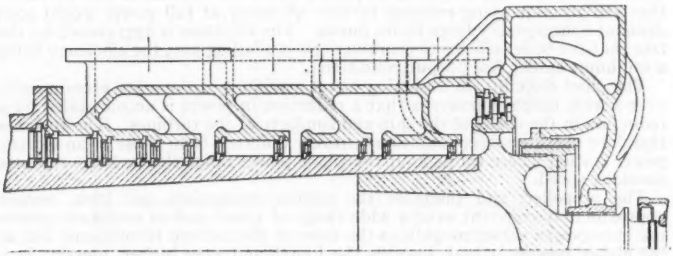


FIGURE 5.—SECTION THROUGH CRUISING TURBINE STAGES.

Nozzle control is, however, only a partial solution of the problem of obtaining economy at low powers, and recourse must be had to inserting more stages or rows of blades in the steam path. This may take the form of a separate cruising turbine arranged to come into action before the high-pressure turbine, or by incorporating additional stages in the high-pressure turbine itself.

Figure 4 shows such a cruising turbine. In general, some form of clutch is provided between it and the high-pressure turbine so that it can be disengaged when the maximum cruising power is exceeded. Originally dog clutches were provided, but they had the disadvantage that they could not be engaged without stopping the propeller shaft. This difficulty has been overcome by using a hydraulic clutch of the Vulcan type, although in some vessels an adaptation of the Synchromesh principle has been used. The problem of alignment of the two rotors under the varying temperature conditions of the two turbines was foreseen and successfully dealt with at the design stage.

In turbines where the cruising stages are incorporated in the high-pressure turbine itself, two or more such stages have been provided, successive bypass valves being opened as power is increased. Care must be taken in designing the area of these valves to arrange that there is always a sufficient pressure drop across the stage to ensure a steam flow through the blades sufficient to keep them cool. Figure 5 shows such an arrangement in a turbine of the reaction type.

Turbines of the impulse type differ somewhat in that interstage valves are fitted between each stage and each diaphragm has a portion of its circumference in which the nozzles are provided with independent externally operated control valves, these nozzles being supplied with boiler steam Figure 6 shows

the detail of such an arrangement for three impulse stages, and in Figure 7 a diagram is given of the steam paths under different conditions.

Figure 8 shows a high-pressure turbine and Figure 9 a low-pressure and an astern turbine

Except in the smallest ships the low-pressure turbine is invariably of the two-flow type. This type of turbine possesses the substantial advantages that it is self-balancing and requires no "dummies"; that on a given diameter of rotor and initial blade height double the area of steam flow is provided as compared with the single-flow turbine; and that both glands are exposed to the same conditions. The last advantage facilitates operational control. The smallness of the astern turbine should be noticed. The full power that it can develop is generally about one-third to one-half of that of the ahead turbines when passing the same amount of steam as the ahead turbines use at their full power.

There have been many excellent and comprehensive papers written on gearing, so I do not propose to deal with this feature other than as regards its effect on the design as a whole. The fact that it is undesirable to exceed a certain unital pressure on the blade fixes the size of the propeller whilst propeller revolutions are governed by the fact that it is also undesirable to

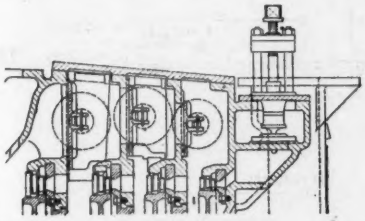


FIGURE 6.—CRUISING STAGES OF BROWN-CURTIS TURBINE.

exceed a certain tip speed of propeller blade. At the other end of the system the turbine designer would like his turbines to run as fast as his experience dictates as being prudent, for in that way he achieves lightness and, if nothing else were involved, the gear ratio that satisfies propeller and turbine conditions would be settled. The gear pinions must, however, be of sufficient diameter to keep the torsional and bending deflections within acceptable limits; and this consideration, in conjuction with the desired gear ratio, determines the size of the main gearwheel.

The position of the propeller is settled by considerations of ship design; and this, in conjunction with the size of the gearwheel and its position in the ship, determines the rake of the shafting, which must not be too great nor such as to bring the forward end of the turbines too near the armor deck.

Consequently it is often found that the gearwheel that both the propeller and turbine designer would like cannot be accommodated, remembering that there must be sufficient head available beneath the main wheels to allow the oil from the gear sprayers, bearings, etc., to drain from the bottom of the gear cases to the forced lubrication pumps; and once again a compromise must be effected.

Figure 10 shows a typical complete set of warship machinery and in it the cruising, high-pressure, low-pressure, and astern turbines will be noticed, as will the gearing. Figure 11 shows a set of gearing. The pinions are disposed to right and left of the vertical center line of the gearwheel and in the top half of the gear case. This arrangement facilitates construction and also examination of the pinions; but, in those designs in which the condenser is underslung, it is actually dictated by the necessity of having the low-pressure turbine sufficiently high in the ship to accommodate the condenser underneath

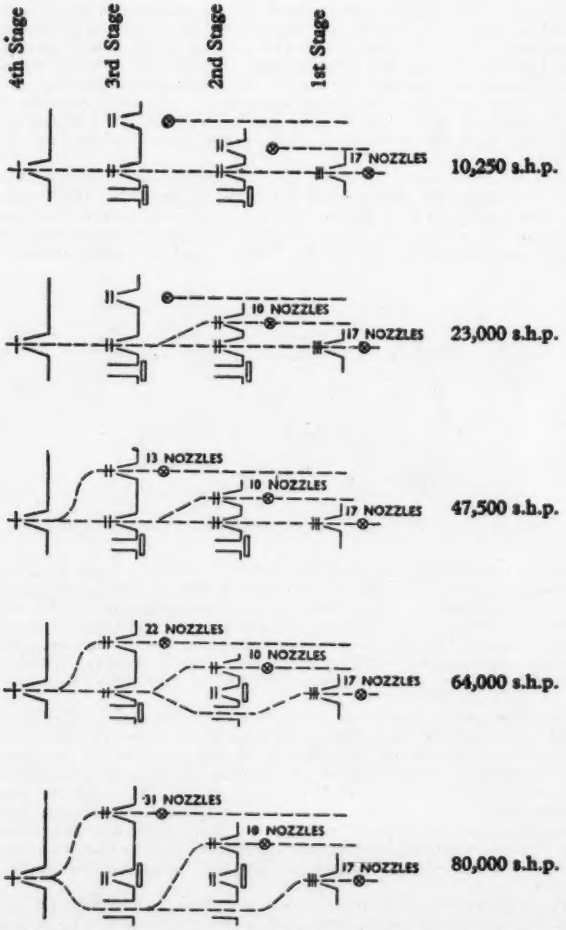


FIGURE 7.—DIAGRAM OF STEAM PATH THROUGH BROWN-CURTIS TURBINE.

it, whilst still allowing sufficient height above the inner bottom of the ship to provide adequate head of condensate above the extraction pump suction. When arranging this, consideration must be given to the effect of list of ship, which list may be intermittent due to rolling or may continue for some little time—as for instance when the ship is turning at high speed under helm, or when she is damaged.

A comparison with high-class Merchant Navy practice is provided by the following example in which the weight of one set of turbines with its associated gearing and condenser weighed some 37 per cent more than the whole machinery installation of a twin-screw naval vessel of the same horsepower as that of the one set, the installation referred to including the propellers, shafting, main engines with associated auxiliaries, pipes, valves—boilers, uptakes, and funnels. The thermal efficiency of the merchant ship plant was, however, nearly 33 per cent higher than that of the naval vessel.

Now let us consider the boiler; this must be light and small and, at the same time, capable of generating steam of uniform quality under very rapid changes of output. In other words, it must also be extremely flexible. It should contain as little water as possible so as to minimize the results of damage; but some water capacity is essential in order that steam may not disappear instantly in the event of momentary failure of the oil fuel supply from any cause. It should preferably be of simple form so as to be readily accessible for examination and cleaning both externally and internally. Its component parts must have a reasonable margin of strength to allow for wear or corrosion, remembering that it may not be possible to insure the absolute purity of feed water under all conditions. Its circulation system must be such that under all conditions of operation, including rolling and listing, local overheating of the tubes can not occur and it must be as efficient as possible consistent with the limitations of space and weight already referred to.

Most boilers in naval vessels today are of the Admiralty type, which is a modified three-drum Yarrow boiler. The water circulation is upwards through the tubes nearest to the furnace and downwards through those more remote from it. Superheat is obtained by inserting hairpin or U-tubes into a space provided by curving the generator tubes each side of it, the ends of these tubes being connected to headers arranged at the front of the boiler and inclined at the same angle as the water tubes. Much theoretical investigation combined with experiment was necessary in order to arrive at the best position of the superheater, to obtain an acceptably uniform degree of superheat at all powers. If it is put too far from the furnace not only will no superheat be produced at low powers but it will in fact act as a condenser, whilst if too near the furnace it will burn out. Figure 12 shows a photograph of such a boiler with the superheater partly withdrawn.

Nowadays practically every auxiliary engine in the larger ships is of the rotary type and uses superheated steam; but before these were available some were of the reciprocating type and used saturated steam. Care must then be taken that the amount of saturated steam withdrawn is not an undue proportion of the total output of the boiler, otherwise there will be an insufficient flow of steam through the superheater and it will be damaged. This possibility can be guarded against by fitting suitably dimensioned choke nozzles in the saturated steam outlet from each boiler, thus limiting the amount that can be withdrawn. This restriction of output has obvious disadvantages and is fortunately no longer necessary, as all auxiliaries in the larger vessels including the evaporators, which are large steam users, now use superheated steam.

The reduction in weight of boilers and in the space occupied by them that has been achieved in the past thirty years by the development of oil-firing has been truly remarkable. A coal-fired battle cruiser built in 1912 had no less than forty-two boilers, whilst today the steam that she required would be very comfortably produced by eight boilers each having about two-and-a-half to three times the heating surface of one of the original boilers. In other words, the evaporation per unit area of heating surface has on the average been doubled, although in extreme cases it has actually been quadrupled.

The reduction in size and weight is greater than the above ratios indicate,

as the employment in more recent boilers of a larger number of smaller diameter tubes is of advantage in this respect.

A rough basis of comparison between boilers can be made by using the ratio of the weight (per hour) of steam which they produce to their own weight. In the more usual type of naval boiler this figure varies from 1.1 to 1.2, although again in extreme cases it may be as much as 2.0.

The weight referred to includes the pressure parts of the boiler, casings to the outlet from the boiler, all mountings (including oil fuel burners, etc.), and water to the working level. It also includes economizer and air heater where fitted. It is not easy to obtain a strictly comparable figure for merchant navy practice, but the maximum may be taken as about 0.4 in the more recent designs

The high rate of forcing, combined with the rapid fluctuation in output demanded of the naval boiler, calls for prompt and accurate control of the feed water supply, and this is effected by the use of automatic feed regulators. If the feed is shut off in a boiler of this kind when it is steaming at full output, the time that elapses before the water disappears from the gauge glass is only 21 seconds and the time to empty the steam drum of water only 68 seconds.

The behavior of boilers, especially those which are highly rated, is somewhat unexpected by those not familiar with their operation. When a greater output is required the increased ebullition results in the density of the fluid in the boiler being considerably reduced, owing to the formation of steam bubbles in the water, and the water level in the boiler rises causing the feed regulator, which is float-controlled, to close. Conversely if the engines are suddenly stopped the water level will fall. Indeed it can be shown that when the change is from full speed to stop, the water level will fall not only below the bottom of the steam drum but some way down the tubes unless feed water is simultaneously pumped into the boiler. Those who have served in ships with reciprocating feed pumps will remember the way in which these pumps suddenly "brought up" when the ship got under weigh rapidly and how they raced away when she stopped. Considerable development has taken place during the past few years in feed regulators of the servo-operated kind and, whereas in the original regulator the feed valve was either wide open or shut, the present ones are all of the "hunting" type and set the feed valve at the correct position to supply the water that is required under any particular set of conditions and gradually shut off the valve as the correct level is approached.

The small water capacity of naval boilers as compared with their large output complicates the problem of feed water supply and occasionally gives rise to the following curious behavior. It must first be explained that in order to increase the efficiency of the installation all auxiliary engines, other than those that have their own condensing plant, exhaust into a closed system, the exhaust steam passing through surface feed heaters where it is condensed. The cycle of events may be taken as commencing when some, or possibly all, the feed regulators open together, which results in a sudden rush of cold feed water through the heaters and the temperature of the feed water entering the boiler drops. This relatively cold water entering the boiler condenses some of the steam bubbles and causes the water level in the boiler to fall. At the same time the passage of cold feed water through the heaters lowers the pressure in the exhaust range, when all the auxiliary engines (not governor- controlled) speed up and pass more steam to the heater. Thus the temperature of the feed water is raised when the reverse series of operations take place. When a point is reached at which the regulators shut off, the fluid in the boiler rapidly increases in volume, causing a sharp rise in water level; the exhaust pressure rises, the auxiliaries slow down, and the cycle is repeated. It seems at times as if the whole system "tunes itself in," and oscillations take place which have a period of 3–4 minutes during which the water level rises and falls in the gauge glass, sometimes disappearing at either end of it. It is a disturbing phenomenon, since the operator can do nothing to check it, but so far it has not produced any untoward results and appears to damp itself out in a short while. It would obviously not occur if the feed water could always be supplied to the boiler at the temperature of the water in the boiler and an

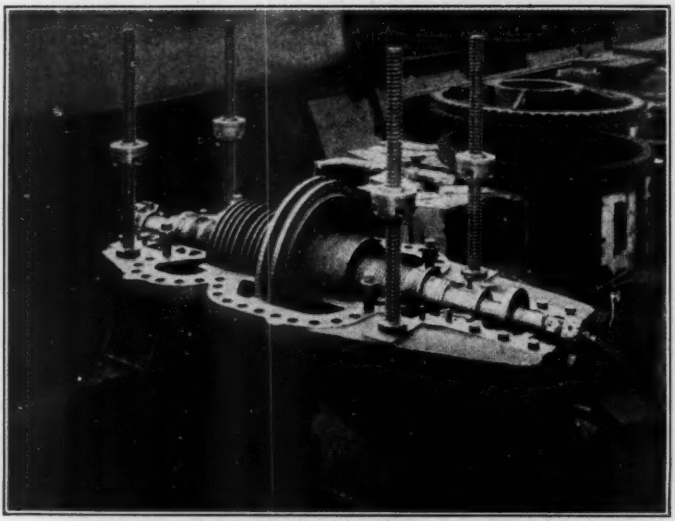


FIGURE 4.—CRUISING TURBINE.

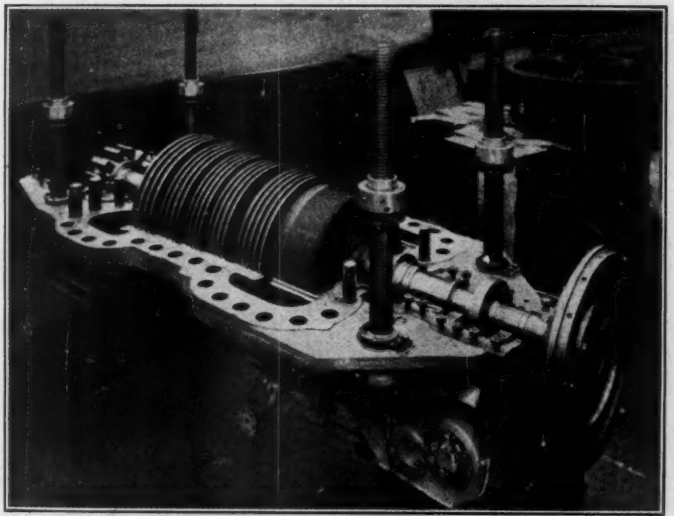


FIGURE 8.—HIGH-PRESSURE TURBINE

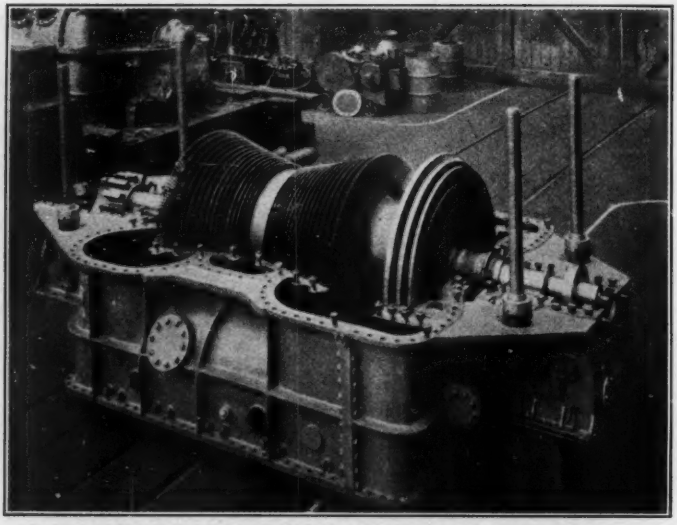


FIGURE 9.—LOW-PRESSURE AND ASTERN TURBINES.

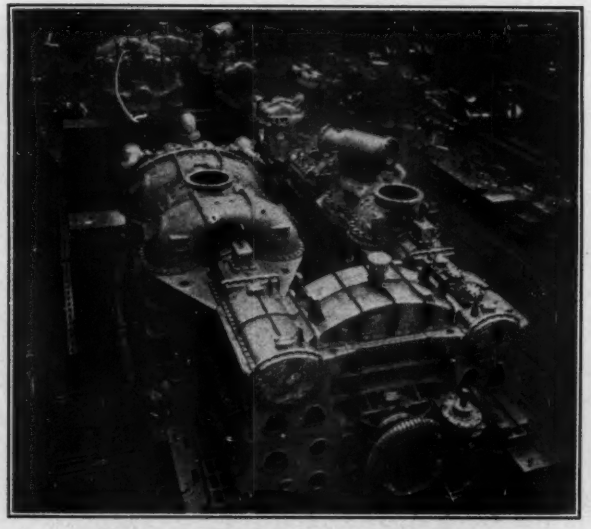


FIGURE 10.—COMPLETE SET OF MACHINERY.

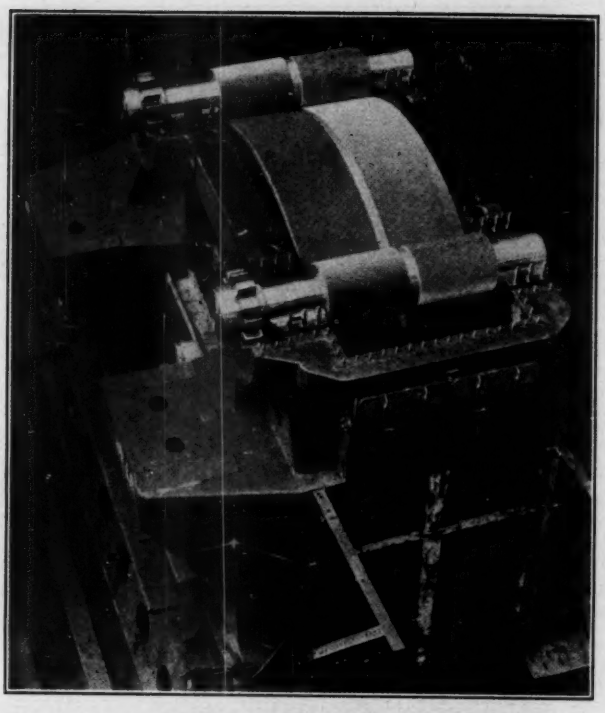


FIGURE 11.—GEARS IN CASE.

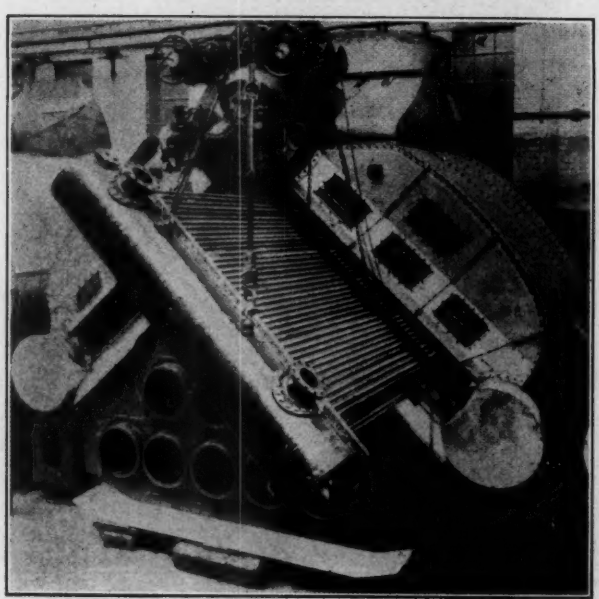


FIGURE 12.—ADMIRALTY BOILER; SUPERHEATER PARTLY WITHDRAWN.

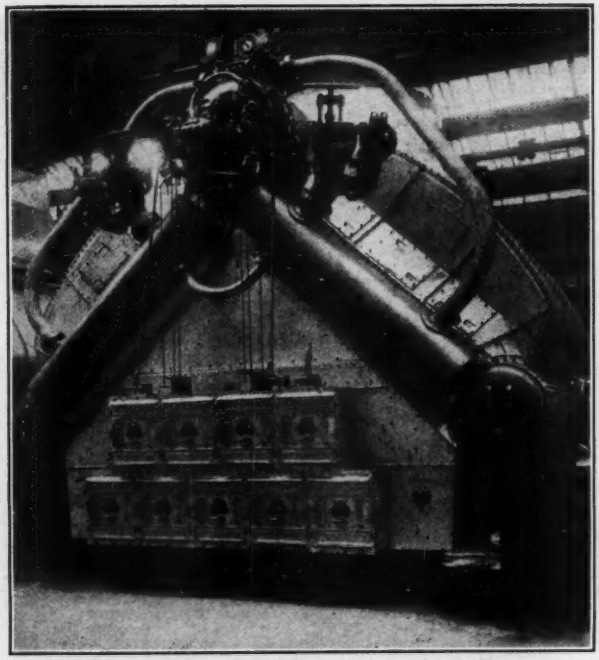


FIGURE 13.—ADMIRALTY BOILER WITH OPEN FRONT.

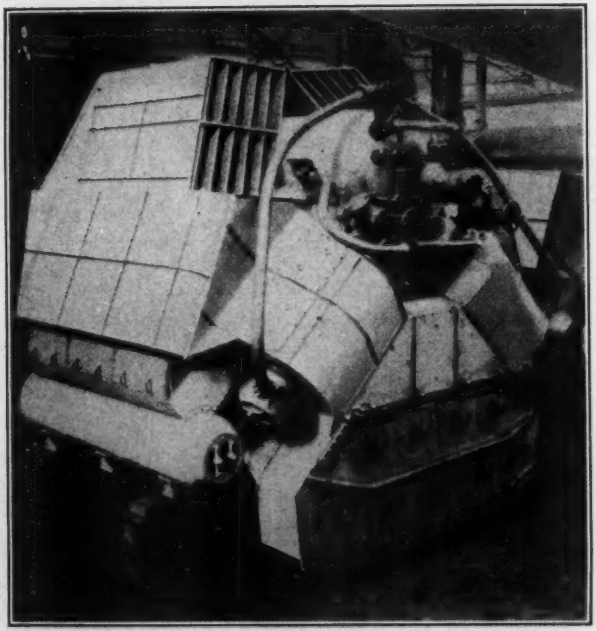


FIGURE 14.—ADMIRALTY BOILER WITH CLOSED FRONT.

approach to this desirable condition is assisted by the use of economizers. To improve the boiler efficiency either economizers or air preheaters are used, an additional advantage being that the lower funnel temperature interferes less with aircraft flying nearby.

An objection to the use of preheaters is that they necessitate a closed air supply to the burners, involving further trunking round the outside of the boiler casing proper, and thus making the external cleaning of the boiler tubes more difficult. Figure 13 shows a destroyer boiler with open front. Figure 14 shows a similar boiler with closed front.

Soot, wetted by rain water coming down the funnels, may cause serious external corrosion of boiler tubes and this is guarded against by fitting catchments in the uptakes and funnels and leading the water collected by them to the boiler room, whence it is pumped overboard.

Somewhat surprising is the fact that no corrosion has occurred in the air heaters that can be attributed to the temperature of the funnel gases being below the dew point when lighting up.

An important part of the boiler installation is the oil fuel burning apparatus. The general arrangement follows the usually accepted marine practice, the only difference being in detail to meet the requirement of a heat release of the order of 280,000 Btu. per cubic foot of combustion chamber volume per hour. This involves the use of burners of individually large output, and even with these the problem is not easy, as the only space available for them is that provided by the front of the boiler.

The wing sprayers must be kept sufficiently far from the fire row tubes to prevent direct impingement of the flame, and the top sprayers must be kept as far below the steam drum as possible. They must be arranged so that the air supply to each sprayer is evenly distributed round it, and this is particularly difficult with the closed fronts which are necessary when preheated air is used. Too little air results in incomplete combustion accompanied by the production of black smoke and, if the shortage is very pronounced, by violent pulsation of the boiler casings. Provided that the other factors such as oil temperature, position of sprayer, cleanliness of sprayer and of combustion tube, are correct, the smoke gradually disappears as the air supply is increased until a "clear funnel" is reached. This clear funnel remains over a wide range of excess air until, in the limit, white smoke, composed mainly of particles of unconsumed oil, is produced. With the latest arrangements oil can be burnt efficiently and smokelessly, average figures giving 14 per cent CO_2 with 12 per cent excess air.

This discussion on air supply brings to mind the valuable aid that can be afforded the designer by model experiments. In a new ship it was desired to make such a big departure from previous practice in the design of the air downtakes, uptakes, and funnels that it was evident that existing data and past experience was likely to be of little value. So models to one-eight scale were made; and so accurately were they constructed, and so well were the laws of similarity applied, that on the full-power trial the pressure actually required was almost exactly that which had been prophesied from the model experiments. I have already made some comparisons with merchant navy boiler practice in the basis of the weight of steam per hour produced by the boiler compared with the weight of the boiler itself; but the following comparative figures may be of interest. They apply to one of our largest merchant ships on the one part and a warship of similar horsepower on the other and include not only the relative boiler weights but those of the turbines, etc., and of the complete installations. The ratios are:—

	Boilers	Turbine gearing and condenser	Complete installation
$\frac{\text{Merchant ship weight}}{\text{Naval ship weight}}$	6·34	3·42	3·71

These figures are illustrative of the different practices that are suited to the needs of the two services. There is no suggestion that either is better than the other, it is just a question of the particular requirements of each and what best meets them.

In the design of merchant navy machinery its first cost and its running cost are obviously factors of great importance and both these can be reduced if space and weight limitations can be relaxed. The comparative figures that I have given show the extent of that relaxation, and will, I hope, serve to inform those who urge that either in the interest of economy or because certain materials are in short supply the Royal Navy should adopt Merchant Navy practice.

It is obviously not possible to produce a picture in which the whole of the machinery installation as fitted in a shore station, a merchant ship, and a naval ship can be shown side by side; but the following information provides a comparison between the different practices so far as boilers are concerned.

In considering the comparative figures given for the various boilers it must be borne in mind that the shore boilers are, in each case, coal-fired, whereas

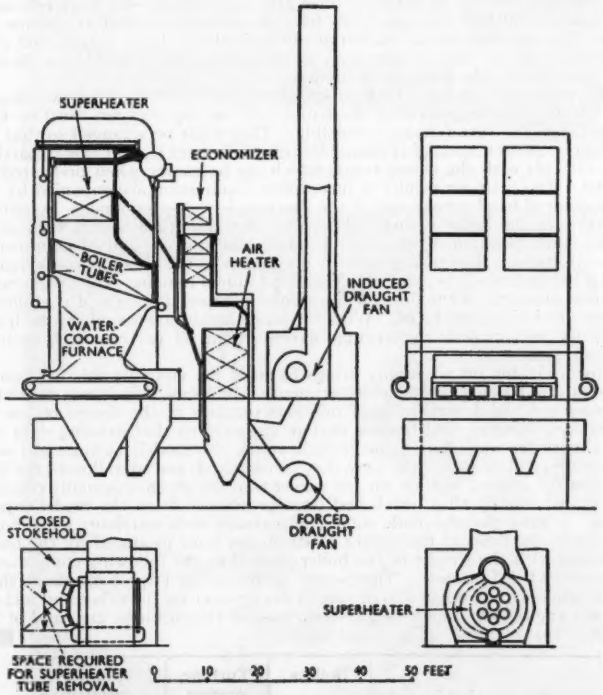


FIGURE 15.—COMPARISON OF LAND BOILERS (BABCOCK AND WILCOX) AND NAVAL BOILERS (BABCOCK-JOHNSON) OF EQUAL OUTPUT- DRAWN TO SAME SCALE.

The top views show the land boiler; and the two smaller views below, the naval type.

both the Merchant and Royal Navy boilers are oil-fired. In particular the figures for efficiency are dependent upon the chemical composition and general quality of the fuel used and there is, perhaps, more variation in this respect in the case of coal than of oil. The weight of the apparatus for coal firing is, naturally, considerably greater than that required for burning oil.

The first example Figure 15 shows, to the same scale, a Babcock and Wilcox power station boiler and one of naval type, each boiler having an output of 150,000 pounds of steam per hour. The weight of the former is approximately 800 tons, excluding ash hoppers, and its efficiency 86.6 per cent on the gross calorific value of the fuel used. The corresponding figures for the latter are 60 tons and 72 per cent. The ratio of the weight of steam per hour to boiler weight is 1.13 in the case of the naval boiler, indicating that it is of normal type. The heat release per cubic foot of combustion chamber space is 29,000 Btu. per hour in the land boiler and 350,000 Btu. per hour in the naval one. The working pressure is 450 pounds per square inch, steam being superheated to 750 degrees F.

Figure 16 shows three Foster-Wheeler boilers for power station, Merchant Navy, and Royal Navy respectively, all having the same capacity of 100,000 pounds of steam per hour. The power station boiler has a superheater depending partly on radiation and partly on convection, proportioned so as to produce steam of approximately constant superheat within its normal working range.

The Merchant Navy boiler has a separately fired superheater of the radiant type and is fitted with economizer and air heater (the latter is not shown). The separately fired superheater provides the great advantage of being able to regulate the degree of superheat, keeping it sensibly constant if desired, or reducing it to very little or cutting it out altogether when going astern.

The other particulars of these boilers which may be of interest are shown in Table 1.

The ratio of steam per hour to boiler weight in the case of the Naval boiler is 1.88, indicating that it is a somewhat extreme design.

Figure 17 shows the three corresponding boilers of the John Thompson type, each of which has a capacity of 100,000 pounds of steam per hour. The power station boiler shown is not fitted with an air heater and is therefore lighter by the weight of this piece of apparatus. Its efficiency is 80 per cent and it is estimated that this would be increased to 84 per cent if an air heater were provided. The Merchant Navy boiler is of the "Integral" type and is fitted with an economizer and an air heater, the latter not being shown.

TABLE 1. WEIGHT AND PERFORMANCE OF LAND AND NAVAL BOILERS (FOSTER-WHEELER)

Type	Land boiler	Merchant Navy boiler	Naval boiler
Weight with water, tons . .	740	145	23·8
Efficiency on gross calorific value, per cent . . .	84	87	76
Heat release per cu. ft. of combustion chamber, B.Th.U. per hour	23,400	68,000	308,000

The Naval boiler is of the Thompson-Lamont type and differs from all the others shown in that it is of the forced circulation variety.

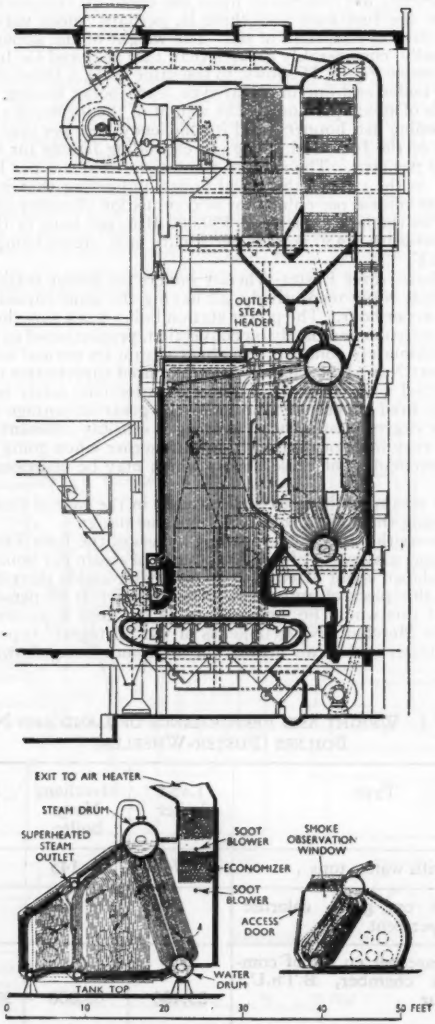


FIGURE 16.—COMPARISON OF LAND AND NAVAL BOILERS (FOSTER-WHEELER OF EQUAL OUTPUT, DRAWN TO SAME SCALE.

The land boiler is shown above; the Merchant Navy type is below on the left and the Royal Navy type on the right.

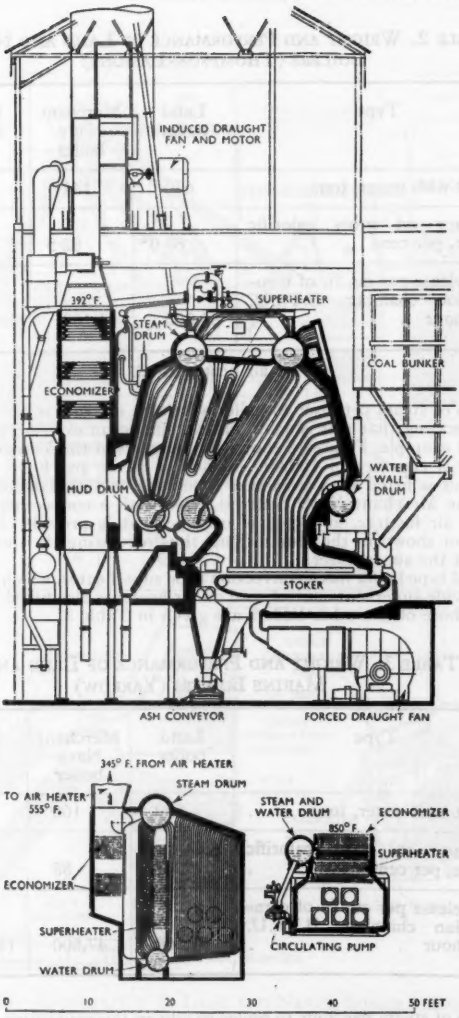


FIGURE 17.—COMPARISON OF LAND AND NAVAL BOILERS (JOHN THOMPSON) OF EQUAL OUTPUT, DRAWN TO SAME SCALE.

The land boiler is shown above; the Merchant Navy type is below on the left; and the Royal Navy type on the right.

Some further particulars are given in Table 2.

TABLE 2. WEIGHT AND PERFORMANCE OF LAND AND NAVAL BOILERS (THOMPSON-LAMONT)

Type	Land boiler	Merchant Navy boiler	Naval boiler
Weight with water, tons . .	670	141	19·5
Efficiency on gross calorific value, per cent . . .	80·0*	82·5	75
Heat release per cu. ft. of combustion chamber, B.Th.U. per hour	34,800	91,100	298,000

* No air heater fitted.

The ratio of steam per hour to boiler weight in the case of the naval boiler is 2.29, indicating that this is again an example of an extreme type.

As a final example, Figure 18 shows three boilers of the Yarrow type, all of which have a capacity of 100,000 pounds of steam per hour. The power station boiler is fitted with covection type superheater, economizer, and air heater. The Merchant Navy boiler is fitted with a convection type superheater and air heaters, and control of superheat is provided by adjusting dampers (not shown in the illustration), thus regulating the quantity of gas flowing past the superheater.

The naval type boiler has a convection type superheater, which is separately fired to provide superheat control, and an air heater is also fitted. Particulars similar to those of the other boilers are given in Table 3.

TABLE 3. WEIGHT AND PERFORMANCE OF LAND AND MARINE BOILERS (YARROW)

Type	Land boiler	Merchant Navy boiler	Naval boiler
Weight with water, tons . .	590	160	24·4
Efficiency on gross calorific value, per cent . . .	84	88	84
Heat release per cu. ft. of combustion chamber, B.Th.U. per hour	31,000	47,800	181,000

The ratio of steam per hour to boiler weight in the naval type boiler shown is 1.83, indicating that once again this is a somewhat extreme design.

These comparisons are made in no critical spirit, but they serve rather to emphasize how different conditions lead to such different designs.

In a paper read by Mr. (now Sir Stanley) Goodall in 1937, the increases in the percentage of the standard displacement of warships absorbed by various

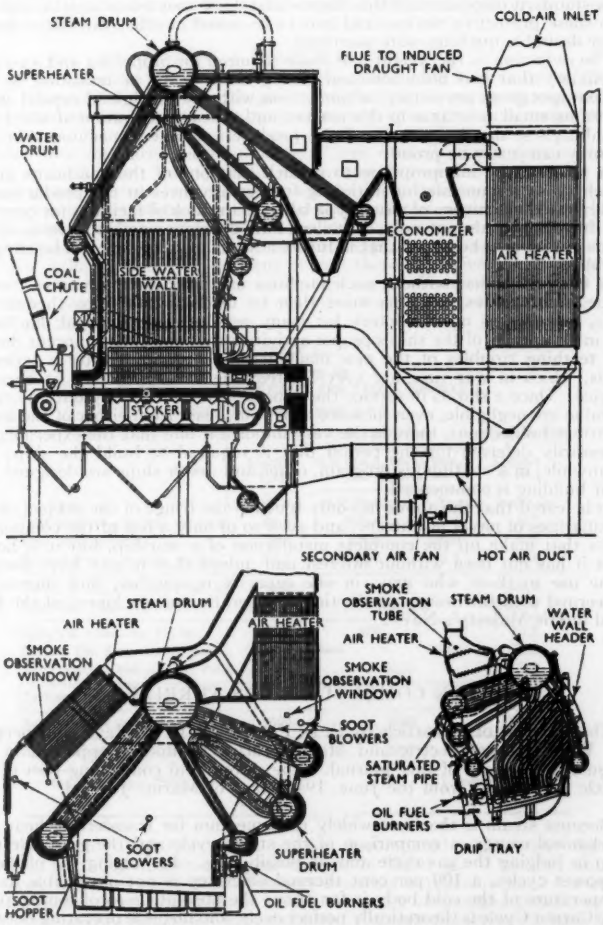


FIGURE 18.—COMPARISON OF LAND AND NAVAL BOILERS (YARROW) OF EQUAL OUTPUT, DRAWN TO SAME SCALE.

The land boiler is shown above; the Merchant Navy type is below on the left; and the Royal Navy type on the right.

forms of protection during the years 1915–36 were discussed. These included anti-torpedo protection, horizontal protection, anti-destroyer and anti-aircraft armament, and in the aggregate involved an increase from 10 per cent in 1915 to 25 per cent in 1936 in the case of capital ships. In a ship of 30,000 tons standard displacement this means that 4500 tons extra weight was put into these protective devices and had to be saved in other directions unless other desirable qualities were sacrificed.

The reduction in the weight and space required for propelling and auxiliary machinery that had been achieved during these years, as indicated by the percentages given previously in connection with the re-engined capital ships, was of no small assistance in this matter, and was a contribution of which the metallurgists, the designers, and the producers of naval machinery in this country can justly be proud.

It may not be inappropriate to remark that one of the handicaps under which those responsible for initiating further advances in the production of highly rated machinery of naval type labor, is the lack of facilities for carrying out full-scale trials not only of boilers but of engines and apparatus of all kinds, and it is to be hoped that in future a more liberal policy will be adopted in this respect.

It is evident that without such facilities only moderate advances can be made, since the experiments must then be made in the ships themselves. This, however, is unsatisfactory for many reasons, the principal one being the interruptions of the ship's proper activities that are likely to occur during the teething troubles of the new machinery. In general, after a series of trials, which in war time are very limited, the vessel proceeds at once on service. Once a ship is in service the opportunities of carrying out any trials of value are negligible, even in peace time; but apart from either of the aforementioned objections, there is the very important one that the experiment is necessarily deferred for the period that is required to build the ship; and meanwhile, in a continuous program, other and newer ships are designed and their building is commenced.

It is feared that the above has only touched the fringe of the subject of the peculiarities of naval machinery, and even so of only a few of the component parts that make up the complete installation of a warship, but it is hoped that it has not been without interest and indeed that it may have been of some use to those who may, in one capacity or another, find themselves concerned with the design, production, or operation of machinery of the kind used in His Majesty's Navy.

THE COMBUSTION GAS TURBINE.

The first part of this article by E. K. Fischer and C. A. Meyer, engineers of the Westinghouse Electric and Manufacturing Company, appeared in the August, 1944, issue of this Journal. The second and concluding part of the article is reprinted from the June, 1944, issue of Marine Journal.

Because steam is the most widely used medium for transferring heat into mechanical energy, a comparison of the steam cycle and the gas cycle may help in judging the gas cycle and its possibilities. In judging the efficiency of power cycles, a 100 per cent thermal efficiency is not obtainable as the temperature of the cold body is far above the absolute zero of temperature. The Carnot Cycle is theoretically perfect cycle, and no cycle operating between the same hot and cold body temperatures can have a better thermal efficiency. The best any cycle can hope to do, when operating under the same conditions, is to attain the Carnot Cycle efficiency. The gas turbine cycle can theoretically give the same efficiency as the Carnot Cycle. This is only theoretically possible because it requires 100 per cent efficiency of the gas turbine and compressor, an infinite number of stages of intercooling and reheating, and a regenerator infinite in size.

The steam cycle theoretically offers the Carnot Cycle efficiency only up to the critical pressure (705.4 degrees F.; 3206 pounds per square inch Abs.). Above this temperature, the gas between the steam cycle and the ideal cycle gradually widens as the temperature is increased. From a purely theoretical standpoint the combustion gas turbine cycle holds forth greater promise of efficiency than the steam cycle.

Gas Cycle Holds Best Promise.

Of more value are the thermal efficiencies obtainable in practical applications. Above 1000 degrees F. the gas cycle efficiency increases approximately three times as fast as the steam cycle efficiency for a given top temperature increase.

This curve compares the best modern large central station steam power plants with the best practical efficiencies expected of the combustion gas turbine power plants. It is expected that the gas turbine cycle efficiency will not be greatly affected by unit size, and this combustion gas cycle curve can also be interpreted as applying to open cycle gas turbine power plants of relatively small capacity. Economic steam plants for capacities around 5000 Kw. seldom exceed 25 per cent overall thermal efficiency. This would indicate a considerable efficiency advantage, in small units, in favor of the gas cycle, provided economics permit designing it for anywhere near its maximum possible efficiency.

TABLE II

COMPARISON OF
COMBUSTION GAS TURBINE AND STEAM TURBINE

5000 Kw Rating—3600 R.P.M.

Item	Open Cycle Gas Turbine: Simple Cycle	Open Cycle Gas Turbine: One reheat, one intercool and regeneration		Steam Turbine: 3 Stages F.W.H.
	Single Cyl.	H.P. Turbine	L.P. Turbine	Single Cyl.
Inlet Pressure—lbs./sq. in. Abs.	88.2	100.4	39.4	465
Inlet Temp.—°F.T.T.	1200	1200	1200	825
Exhaust Press.—lbs./sq. in. Abs.	14.7	39.4	15.1	3/4
Steam or Gas rate—lbs./Kw. hr.	102	56		10.5
Full Load Flow—lbs./hr.	510,000	280,000		52,500
Inlet Vol. Flow—cu. ft./sec.	990	475		24.8
Exhaust Vol. Flow—cu. ft./sec.	3900		3250	6120
Ratio (Exhaust Vol. ÷ Inlet Vol.)	3.95	6.85		250
Turbine Floor Area—sq. ft.	160	150	155	100†
Net Generator Output—KW	5000	5000		5000
Approx. Plant Thermal Eff.—%	18.4*	29.4*		23.6

* Inlet Air Temp. 70°F.

† Turbine Room Only.

A 3600 Rpm., 5000 Kw. steam turbine is compared with a 3600 Rpm., 5000 Kw. net output open cycle gas turbine in Table II. The first column compares the simple cycle without reheat, intercool or regeneration. Columns 2 and 3 make the comparison for a 5000 Kw. net output gas turbine cycle using 1 intercool, 1 reheat and regeneration. Column 4 compares a 5000 Kw. steam turbine. This comparison indicates the low pressure in the open cycle gas turbine as compared with the steam cycle. As a result, the energy per pound of gas is small and the flow of gas is very large; 510,000 pounds per hour for the simple cycle gas turbine as compared to 52,500 pounds per hour for steam to the steam turbine. The large flow and low pressure to the inlet of the gas turbine gives a large volume flow which means the piping and blading of the gas turbine inlet are large compared to the steam turbine.

The ratio of exhaust to inlet volume is small for the gas turbine. This makes a balanced blade path unlike that of the steam turbine, which in this case must handle an exhaust volume 250 times as great as the inlet volume. The large blade dimensions limits the maximum net output rating of the open cycle, single flow, combustion gas turbine to approximately 7500 Kw. However, the injection of liquids presents a theoretical possibility considerably extending this limit.

Closed Cycle for Large Units.

The closed cycle offers a method of increasing the maximum capacity of the open cycle. The volume of the working gas is inversely proportional to the absolute pressure. If the pressure is multiplied by 10, the size is divided by 10. In the closed cycle, the circulating working gas is at a relatively high pressure and reduces the physical size of compressor and turbine. To reduce the temperature of the gases before they enter the compressor, cooling water is required in the closed cycle. The heat exchanger in which the gas is cooled before it enters the compressor is called a gas precooler. The amount of heat given up to the cooling water is equivalent to that removed in the condenser of a steam unit of equal capacity. The quantity of cooling water required will be less, as a higher water temperature rise is permissible.

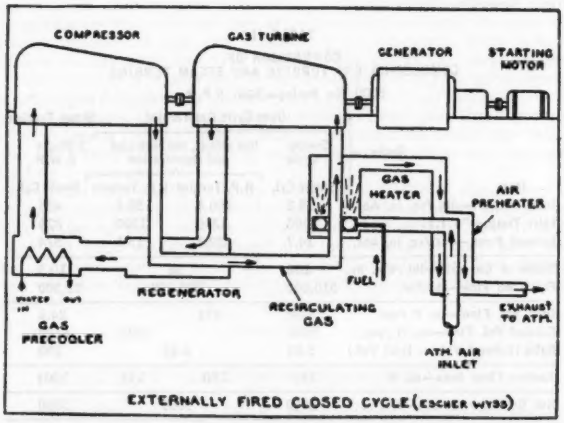


Figure 11.—Escher Wyss Externally Fired Closed Cycle.

The closed cycle in Figure 11 is the one used by Escher Wyss. This is an externally fired cycle in which the products of combustion do not pass through the gas turbine and compressor. The working gas (air, hydrogen or other medium) operates at a relatively high pressure in a closed, recirculating circuit. In the gas heater, the heat from the products of combustion is transferred to the working gas which then expands in the turbine to a lower pressure. The gas heater in this cycle corresponds to the steam boiler in the steam cycle. For a practical efficiency, it will be larger than the modern steam boiler because gases are on both sides of the gas heater. This cycle is very similar to the steam cycle, except that the working fluid does not undergo a change of state. As this closed cycle keeps the products of combustion out of the turbine and compressor circuit, the problem of using coal as a fuel should be much simpler of solution than in those cycles which circulate the products of combustion.

Hydrogen Gas Advantages.

In the closed cycle, the compressor inlet pressure will be maintained at approximately 150 pounds per square inch with a discharge pressure of some 600 pounds per square inch. This high pressure greatly reduces the size of turbine and compressor and should permit maximum ratings to be built approaching those in the steam cycle. Some gas other than air will probably be used as the working medium in the externally fired closed cycle. Hydrogen, for example, has properties which make it far superior to air for this application. The density of hydrogen is 1/14 that of air, the specific heat is 14 times that of air, and the thermal conductivity is 6.8 times that of air.

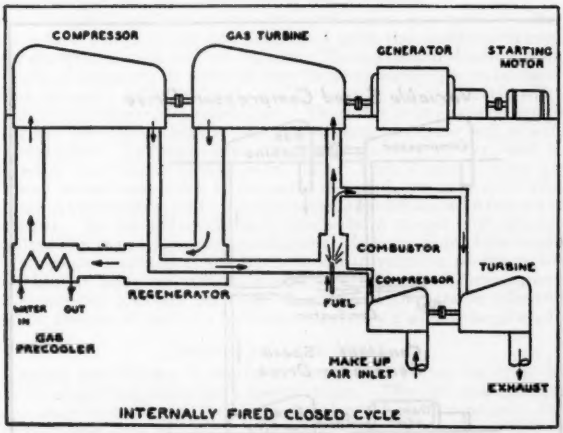


Figure 12.—Proposed Internally Fired Closed Cycle.

New Cycle Studied.

A closed cycle system under development by Westinghouse, is shown in Figure 12. In this closed cycle a separate gas turbine and compressor are used to pump up the cycle on which the main gas turbine and compressor operate. High pressure of around 600 pounds per square inch may be used with a compressor inlet pressure of around 150 pounds per square inch. The main gas turbine and compressor would be small as they operate at high pressure. This cycle is internally fired, the products of combustion passing through the gas turbines and main compressor. Enough make up air is continually supplied to maintain pressure and support combustion. It is supplied by a compressor which is driven by a second gas turbine. This cycle avoids the large gas heater required by the Escher Wyss cycle; but requires an extra gas turbine and compressor to pump-up the system. Solid matter from the fuel must be removed.

In the closed cycles shown in Figures 11 and 12, reheating and intercooling are not illustrated. However, they offer the same advantage in the closed cycle that they offer in the open cycle. The biggest single additional problem in the closed cycle is a method of building practical heat exchangers. The problem is further complicated by the fact that the gases will carry foreign matter from combustion which may both corrode and erode the exchanger and reduce the heat transfer rate by depositing foreign material on the transfer surface.

Gas Turbine Control.

Combustion gas turbine control can be simple and reliable, consisting only of control of the gas temperature by controlling the rate of fuel supply. Governing valves, such as used in steam turbine control, are not needed. Efficient partial load performance can be obtained by using two turbines: one variable speed turbine driving a compressor; plus a constant speed turbine driving a generator. See Figure 13. The use of regenerators, reheaters, and intercoolers in addition to improving the full load economy. Have an even greater effect in improving the partial load economy. In the closed cycle, by reducing the gas pressure as the load is reduced, practically full load efficiency can be maintained at partial loads.

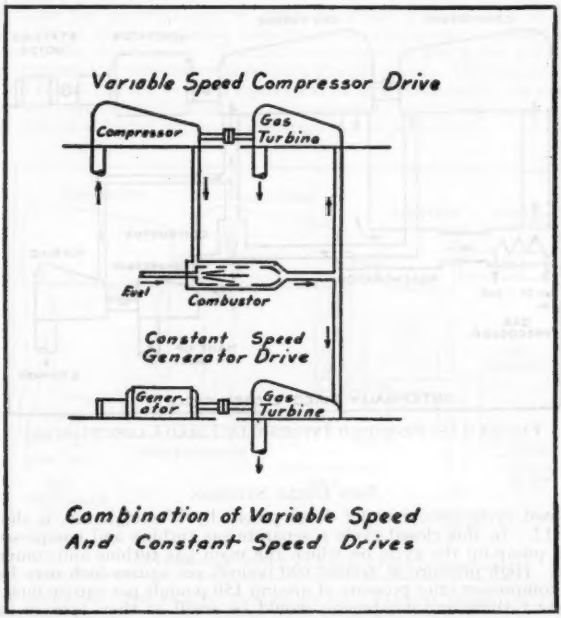


Figure 13.—Possible Method of Gas Turbine Control Combining Variable and Constant Speed Drives.

Possible Gas Turbine Applications.

The possible applications of the gas turbine are many. Ranging from a simple open cycle for small capacities to a closed cycle for very large ratings it offers wide possibilities. For example:

Locomotives. The simple open gas cycle requires no water. It has low weight and small space requirements combined with simplicity. With an efficiency of 20 per cent at 1200 degrees F. and the expected low maintenance of turbine drives, it should prove a good power plant for a locomotive. The inability of the gas turbine to operate in reverse makes either electric drive or the development of a satisfactory reversing gear necessary for this application.

Airplanes. For relatively small power outputs, such as required by the airplane industry, the gas turbine operating at high speed and high temperature to obtain maximum rating per unit weight of material, has real possibilities.

Ship Drives. The combustion gas cycle offers efficiencies equal to the best modern marine steam power plants, which have overall efficiencies of approximately 25 per cent. Weight and space requirements of equipment are a real factor in this application. The gas cycle eliminates the steam generator and steam condenser. The advantage in weight and space undoubtedly will favor the gas cycle, despite the gas compressor, the regenerator heat exchanger, and the large gas turbine. The requirement of astern operation in marine service handicaps the gas turbine, except as an electric drive.

Power. General application of the combustion gas cycle in the power generation field will probably not take place until the problems in connection with the use of coal as a fuel are solved. The successful development of the closed cycle is necessary if units of very large capacity are to be built. The maximum capacity for which units can be built in the open cycle will include the majority of industrial applications. There are many special applications in the power generation field in which the open gas cycle will possibly find early application. Examples are: emergency standby service, and low first cost units on the ends of transmission lines. Here a simple open gas cycle offers many advantages, such as no water required, low first cost, simplicity, small space requirements, and virtually automatic station with few attendants.

Processing. In the industrial field where both power and process steam are required the gas turbine has possibilities. It fits well in those applications where the steam required is relatively small in relation to the power load. This is different from the extraction steam turbine where large quantities of process steam per Kw. are necessary to attain a comparably efficient cycle. Here again, the use of coal as a fuel is necessary for a wide general application.

Many Problems Ahead.

In drawing conclusions, it should be remembered that the cycle has only had practical application in very special cases. The full possibilities of any cycle can only be evaluated from successful proof of its economy, first cost, maintenance cost and reliability. The addition of elements which improve the fuel economy, and arrangements of the cycle for large capacities, are obtained at a sacrifice in simplicity and at a price. The deevlopment of the best system is expected to be costly in time and money.

Metallurgy plays an important part in the gas cycle as the efficiency increases rapidly with increase in top temperature. To obtain materials suitable for operation at higher temperatures the metallurgists are looking at materials similar to the non-forgable and non-machinable tool steels. The method of forming these alloys to shape, such as precision casting to size, may revolutionize accepted methods of manufacture. To apply such materials their additional first cost and manufacturing cost must be justified. Any application of such materials must be preceded by careful tests. For heavy duty apparatus these tests must extend over long periods before the designer can use them with safety. Careful differentiation between applications as to required length of life of apparatus is necessary. The fact that a piece of equipment is operated at 1800 degrees F. for a life of a few hours does not mean temperatures of that order can be used for heavy duty applications.

Present developments of the gas turbine are limited to the use of relatively high grade fuel oils. This one factor is a serious handicap to the gas cycle. There is considerable evidence that oil is being used at a greater rate than new supplies are being found. So in the post-war period necessity may dictate a prime mover which can use coal as a fuel. The gas cycle is definitely limited in application until such time as the problems in connection with the burning of low grade oil and coal are successfully solved.

Present research efforts are being expended in developments for the National Defense. In the post-war era industry will have access to the developments in the gas turbine field and to developments in high temperature materials.

These developments may considerably change present thinking. It is safe to predict that general application of the gas cycle must wait until the post-war era.

It will be wise to watch the developments of the early installations before attempting to make widespread applications. At present, conclusions as to the ultimate possibilities of the gas cycle are little more than good guesses. The gas turbine art must advance beyond its present early development stages, before it can be judged with assurance. However, undoubtedly it will find real usefulness in a large number of fields, possibly complementing rather than competing with the steam turbine. Just how and where the gas turbine will be applied, only time will tell.

MISCELLANEA OF INTEREST TO NAVAL ENGINEERS.

THE RAPIDLY EXPANDING FIELD OF X-RAYS.—The following paper was read at the Medal Day Meeting of The Franklin Institute, on April 19, 1944, by William David Coolidge, Vice-President and Director of Research, of the General Electric Company, recipient of an award from the Institute. The paper was published in the June, 1944, Journal of The Franklin Institute.

In what follows I shall deal briefly with a single field in which our laboratory is actively engaged and concerning which we are reasonably free to talk. I say "reasonably" because even here certain phases of the work are under secrecy orders.

I refer to the x-rays and their rapidly expanding field of usefulness. This expansion is due to many causes and to the constructive efforts of many men. It is due, on the one hand, to improvements in x-ray generating equipment and, on the other, to the step-by-step process of learning what this form of radiation can help us to accomplish.

To speak of the x-ray generating equipment and, first of all, the tube: the present type, the hot-cathode high-vacuum tube, in its original commercial form was limited to voltages not in excess of about a hundred and forty thousand, while later models were developed for voltages up to four hundred thousand. It was difficult to go much beyond this until we adopted the expedient of grading the potential along the tube by means of a multiplicity of tubular accelerating electrodes. In this form the tube apparently can be designed for voltages as high as can be produced. Many tubes of this multi-section type are operating at a million volts and similar sealed-off tubes for two million volts have already been developed. These last produce x-rays like the gamma rays from radium

This increase in the voltage at which the tube can be operated extends the available x-ray spectrum, for the x-ray wave-length is simply inversely proportional to the voltage, and the penetrating power of the rays increases rapidly with decreasing wave-length.

By a new method we are now able further to extend the available range of x-rays up to those corresponding to a *hundred* million volts.

In the other direction, by the use of a beryllium window in the tube, we are able to render available x-rays of useful intensity at voltages as *low* as a thousand.

In these ways the available x-ray spectrum has been extended until it now covers more than sixteen octaves.

Our eyes respond *directly* to a light spectrum embracing only one octave. By *indirection* they may be made to respond to any portion of the entire x-ray spectrum; hence we may say that, through the x-rays, the range of our vision has been extended by sixteen octaves.

Various forms of high voltage source have been used for operating the x-ray tube, including the induction coil, various forms of static machine, the high frequency coil, the transformer with and without rectifying device and, last of all, Mr. Westendorp's low frequency resonance transformer. I have

omitted reference to one other source, namely, that source tapped by Dr. Franklin with his kite. Brasch and Lange working on Mt. Generoso in Switzerland attempted to make use of this. They found it potent but inconvenient. The ordinary transformer is now generally employed up to 400,000 volts, while for the range from a couple of hundred thousand to a couple of million volts the low frequency resonance transformer is especially well suited.

For the generation of very high voltages, the use of a suitable compressed gas instead of oil as the insulating medium has resulted in a substantial reduction in size and weight of equipment.

Improvements in design resulting in the elimination of danger to the operator from both electric shock and exposure to the rays have also had much to do with their increased use. Reduction of the latter hazard—exposure to the rays—has been greatly facilitated through the development by the physicist of satisfactory devices for *measuring* x-ray intensity and through the determination by the medical profession of the x-ray dosage to which a human being can safely be exposed. This last is very important because no matter how thick the lead or other protective material used, there is always some radiation getting through.

Apparatus developments resulting in easy *portability* have made it possible, where necessary, to bring the x-rays to the patient. For war use, self-contained portable equipment, complete with its own power plant, makes possible x-ray work close to the front.

To speak now of some of the effects of the war needs:

In medical diagnostic work, a radiograph of the chest has called for a large film, usually 14 by 17 inches. The x-ray examination of the chests of all of our recruits by this method would have called for a tremendous amount of photographic film. This situation has been greatly eased by substituting for the radiograph a photograph, made with a small camera, of the x-ray shadow on the fluoroscopic screen, thus reducing to a tenth, or even less, the amount of film required. This procedure is resulting in a much wider use of x-ray diagnosis than would otherwise be practicable and will carry over into peacetime, especially for the frequent chest examination of school children.

In industry, the war has greatly increased the fluoroscopic use of the x-rays. Without such non-destructive inspection, small aluminum and zinc die-castings which looked all right on the surface would be machined and might then show defects, necessitating rejection with the loss of whatever man-hours of labor had gone into them. Such parts are now put on an endless belt and, as they pass through the x-ray beam, observed fluoroscopically. Prior to the use of x-ray examination, rejections after machining have sometimes run very high, even in excess of fifty per cent. The x-ray inspection not only protects the *user* of the castings from loss of labor but also enables the *producer* of the castings to maintain a constant check upon his output with the result that he doesn't produce bad castings.

Some industrial fluoroscopic examinations have been made *entirely automatic* by substituting for the human eye a photo-cell, and by using an amplifier for the electric current produced and electro-mechanical means for either marking the defective parts or separating them from the good ones. In this way it has been possible, for example, to inspect hand grenade fuses at the rate of 4000 per hour.

Million volt x-rays are today playing an important role in industry in the radiographic examination of various kinds of war material including welds and steel castings as thick as eight inches. Their main advantage over lower voltage rays consists in their greater penetrating power and the correspondingly shorter exposure time required. As an illustration of this, a radiograph through five inches of steel, made with a certain four hundred thousand volt outfit at a certain distance and with a certain film, requires *three and a half hours* while, with a one million volt outfit, only *two minutes* is needed.

Furthermore, the higher voltage x-rays helpfully reduce the contrast in the radiograph and so make possible the study from a single film of a specimen of widely varying thickness.

As an illustration of this I may cite the case of a magnesium alloy casting

varying in thickness from a quarter of an inch up to twelve inches. A single million volt radiograph gave the information throughout this thickness range which, with two hundred thousand volt x-rays, was to be had only by making about nineteen radiographs. Another great advantage in the use of such penetrating radiation consists in the fact that with increasing voltage the secondary x-radiation produced in the object is scattered less and less in direction and hence causes less and less blurring of the shadow picture.

For industrial use, automatic machines have been developed for positioning parts to be radiographed and for making the exposures, and other automatic machines have been produced for processing the film.

The x-rays may be used as a thickness gage, without the necessity of making mechanical contact with the work, as, for example, in the rolling of sheet steel, where the sheet metal may be coming through the mill at the rate of over twenty miles per hour and at temperatures up to 2000 degrees F. and where, through adjustment of the mill, the thickness of the sheet must be kept within certain narrow limits. With an x-ray outfit below and an ionization chamber or other x-ray intensity measuring device above the sheet, it becomes possible through the measurement of x-ray transmission to have a constant indication of thickness and, if desired, to have the x-rays themselves control the mill so as to maintain automatically a constant thickness of the steel sheet.

Starting with the various *medical* uses, we have seen the field of application of the x-rays extend in many other directions. To mention a few of these:

They may serve as a means of chemical analysis.

They are used to measure the distances between the atoms in a crystal and so to determine crystalline structure.

In cutting quartz crystals for radio-frequency control, x-rays are used to determine the orientation of the axes, in case the natural faces, which could otherwise serve as a guide, are lacking.

Through the measurement of the spacing of atoms they make it possible to show existing strains in structural materials—strains which if not removed might lead to structural failure.

They reveal hidden flaws in structural materials.

They may be used for the inspection of finished apparatus to check up on alignment of parts, as in metal vacuum tubes, or to show that no essential part has been omitted as in fuses.

They are used for the inspection of automobile tires.

They are used fluoroscopically for the inspection of foodstuffs such as candy, diced vegetables for canning, shelled nuts and dried fruits to guard against the inclusion of foreign bodies. They have also been used to judge the condition of fresh citrus fruits.

The very *low voltage* x-rays are used in microradiography as a means of studying the composition of very thin metallic specimens and to reveal structure in delicate botanical and zoological tissues.

Lastly, and to the physicist most important, the x-rays have, perhaps more than any other single agency, advanced our fundamental knowledge of the structure of matter

The first world war greatly accelerated the *medical* use of the x-rays and this war is doing the same for the *industrial* field.

I have said but little about the medical use, concentrating mainly on the *more recent* industrial applications.

Of the sixteen octaves of the spectrum now available for study, the upper five octaves have scarcely been explored at all. Their fields of usefulness remain to be determined. It is known that as voltage is raised above a few million, penetrating power reaches a maximum, differing for different materials, and then declines, but aside from penetrating power it seems certain that the higher energies obtainable at the higher voltages will either open new fields of utility, which are as yet hardly more than fields of speculation, or will lead to an increase in our knowledge of the structure of matter, thereby in either case further expanding the field of usefulness of the x-rays.

BOOK NOTICES.

PRINCIPLES OF POWDER METALLURGY, BY FRANZ SKAUPY. PUBLISHED BY PHILOSOPHICAL LIBRARY, NEW YORK. 80 pp. $3.00.

ILLUSTRATED TECHNICAL DICTIONARY. Containing standard technical definitions of current terms in the applied sciences, graphic and industrial arts, and mechanical trades; including air navigation, meteorology, shipbuilding, synthetics and plastics; with illustrations, technical data, and intraconversion tables. EDITED BY MAXIM NEWMARK, AUTHOR OF A DICTIONARY OF SCIENCE AND TECHNOLOGY IN ENGLISH, FRENCH, GERMAN, SPANISH. PUBLISHED BY THE PHILOSOPHICAL LIBRARY, NEW YORK. 352 pp. $5.00.

MARINE ENGINEERING, VOLUME TWO. Written by a group of authorities. EDITOR—HERBERT LEE SEWARD, PH.B., M.E. PROFESSOR OF MECHANICAL AND MARINE ENGINEERING, YALE UNIVERSITY. PUBLISHED BY THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS, 29 WEST 39TH STREET, NEW YORK 18, N. Y.

ASSOCIATION NOTES

The Annual Meeting of the Society was held in Washington, D. C., on Tuesday 3 October, 1944.

The following were nominated for offices for the calendar year 1945:

President:

Rear Admiral E. W. MILLS, U. S. N.

Secretary-Treasurer:

Commander R. T. SUTHERLAND, JR., U. S. N.
Captain F. W. WALTON, U. S. N.

Members of Council:

Rear Admiral ALEX M. CHARLTON, U. S. N.
Captain P. F. LEE, U. S. N.
Captain S. N. PYNE, U. S. N.
Captain H. G. RICKOVER, U. S. N.
Captain E. E. ROTH, U. S. N.
Captain MAX SCHREINER, U. S. N.
Captain H. C. SEXTON, U. S. N.
Captain L. F. SMALL, U. S. N.
Rear Admiral F. J. WILLE, U. S. N.
Commander P. V. COLMAR, U. S. C. G.
Commander HARRY E. DAVIS, JR., U. S. C. G.
Lieut. Commander LOUIS HAUSMANN, U. S. N. R.
Lieut. Commander E. G. KINTNER, JR., U. S. N. R.
Mr. W. H. BAILEY.
Mr. JOHN BLIZARD.
Mr. FRED SELLS.
Mr. C. R. WALLER.
Mr. G. B. WARREN.

Ballots have been distributed. Polls close at 3:30 p. m., 26 December, 1944.

BANQUET 1945. It has been decided to hold a banquet on Friday, April 20, 1945, at the Statler Hotel, 16th and K Streets, N. W., Washington, D. C. Details will be announced as soon as arrangements have been completed.

Prize Article by Naval Reserve Officer. Articles are invited from Reserve Officers on the general subject of "Engineering in the Navy as Seen by an Active Reserve Officer," to be submitted up to 31 December, 1945. Prizes of $500, $250 and $100, respectively, may be awarded for articles considered by the Council to warrant such award. The prize will be in addition to the regular payment for articles published.

MEMBERSHIP.

The following have joined the Society since the publication of the August, 1944, JOURNAL:

NAVAL.

Adams, James F., Jr., Lieut., U. S. N. R.

Armstrong, Charles V., Lieut., U. S. C. G. R., Coast Guard Headquarters, 1300 E St., N. W., Washington, D. C.

Barriger, Melvin Howard, Ensign, U. S. N. R., 2917 North Thomas Road, Fairgrove, Mich.

Bartoszek, Stanly G., Lieut., U. S. N. R., Naval Armory, Chicago 7, Ill.

Brennan, J. M., Machinist, U. S. N.

Byrne, John W., Ensign, U. S. N. R., 3306 Thelma Avenue, Los Angeles, Calif.

Christensen, Robert P., Chief Machinist, U. S. N.

Christina, Richard E., Electrician, U. S. N., 440 Clifton St., Los Angeles, Calif.

Creighton, William H., Lieut. Commander, U. S. N. R., 3016 Tilden St., N. W., Washington, D. C.

Crue, George E., Electrician, U. S. N.

Dixon, George T., Lieut., U. S. N. R., Office of Resident Inspector of Naval Material, General Electric Company, Pittsfield, Mass.

Fitz Gerald, S. A., Electrician, U. S. N.

Flaherty, Edward Bromly, Commander, U. S. N. R.

Haack, Albert F., Ensign, U. S. N.

Haeberle, Frederick E., Captain, U. S. N., Bureau of Ships, Navy Department.

Hardesty, T. W., Lieut., U. S. N., Room 200 Commerce Building, 744 N. 4th St., Milwaukee, Wis.

Heald, J. H., Ensign, U. S. N.

Hirth, Andrew P., Lieut. Commander, U. S. N. R., 10 Bitter Sweet Lane, Valley Stream, L. I., N. Y.

Hobbs, George S., Ensign, U. S. N. R.

Holz, Charles Henry, Lieut., U. S. N. R.

Houston, Joseph A., Lieut. Commander, U. S. N.

Joers, Joseph E., Lieut. j. g., U. S. N. R.

Laird, H. C., Captain, U. S. N., Ret., Lakewood, Norfolk, Va.

McEwan, James Guthrie, Lieut., U. S. N. R.

Martin, Homer A., Machinist, U. S. N.

Mastrobattista, A., Lieut., U. S. N. R., Resident Inspector of Naval Material, General Electric Co., Pittsfield, Mass.

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Mumma, Albert G., Commander, U. S. N.

Oest, Harry A., Lieut. Commander, U. S. C. G., Coast Guard Headquarters, Washington 25, D. C.

Orr, John J., Lieut., U. S. N. R., Asst. Inspector of Surface Ships, 2115 Chestnut St., Philadelphia 3, Pa.

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Rupp, Lewis A., Lieut. Commander, U. S. N., David W. Taylor Model Basin, Washington, D. C.

Seaberg, Arthur A., Lieut. Commander, U. S. N. R.

Shean, Thomas L., Lieut. Commander, U. S. N. R., Office of Industrial Manager, 9th Naval District, 175 West Jackson St., Chicago 4, Ill.

Smith, Luther M., Lieut., j. g., U. S. N. R.

Snyder, Asa Edward, Ensign, U. S. N. R.

Spike, Torrance G., Lieut., j. g., U. S. N.

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Ulrich, Herman Grant, Ensign, U. S. N.

Vernon, Eugene D., Lieut., U. S. N. R., 1235 Bush St., San Francisco, Calif.

Walter, Robert Shirly, Ensign, U. S. N. R.

Wilhelm, Oscar W., Ensign, U. S. N. R., Production Division, Navy Yard, Puget Sound, Wash.

Wilkin, Robert W., Lieut., U. S. N. R., 195 Beresford Avenue, Highland Park, Mich.

CIVIL.

MacIntyre, James R., Asst. Chief Engineer and Asst. Director of Research, Nash Engineering Co., South Norwalk, Conn.

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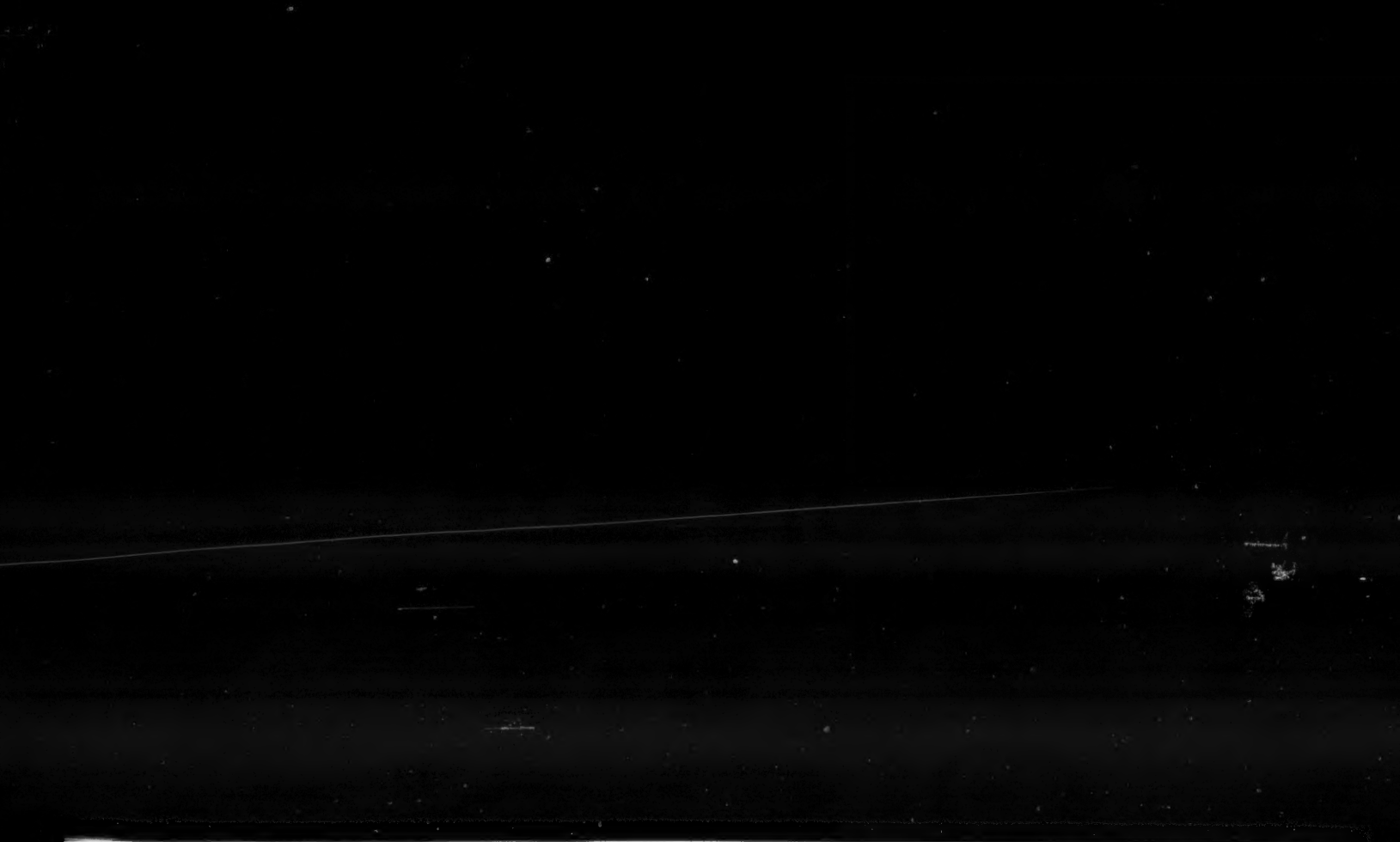
ASSOCIATE.

Hirth, Raymond C., Principal Draftsman, Ind. Dept., Design Section, Hull Mech. Group, Navy Yard, Philadelphia, Mail, Windsor Essex Inn, Narberth, Pa.

Murdock, O. V., Ensign, U. S. M. S., 6 Sebago Drive, Apt. M., Portsmouth, Va.

Robertson, A. D., Asst. Mgr. Norwood Works, Allis Chalmers Mfg. Co., 4262 Forest Ave., Norwood, Ohio.